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The final design demonstrates that commercially available components can satisfactorily be employed in the adaptation of the GSD-10 to the AAV-7A1, but the desire to minimize system weight resulted in the integration of the primary and secondary differentials into one housing.

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SEACO, INCORPORATED

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February 1987

ABSTRACT

An engineering design study was conducted for the adaptation, assembly and installation of the Gleasman Steer Drive (GSD-10) system for the U.S. Marine Corps AAV-7A1 tracked amphibious vehicle. The GSD-10 is an innovative solution to the problem of tracked vehicle steering and is composed primarily of commercially available, off-the-shelf components. Analyses included the collection of relevant commercial drive train component data, a detailed layout of a vehicle drive train system, a transmission and steering control system study, weight and balance estimates, and a performance analysis of the proposed system.

The final design demonstrates that commercially available components can satisfactorily be employed in the adaptation of the GSD-10 to the AAV-7A1, but the desire to minimize system weight resulted in the integration of the primary and secondary differentials into one housing.

The proposed system has been designed to meet the prescribed gradability and top speed requirements. Components for the steering control system have been sized so that complete steering control is available at all vehicle speeds.

Confinement of the power train to the existing AAV-7A1 engine compartment yields a non-optimal configuration of the transmission/steer drive system. While the system was arranged so that no detrimental impact on center-of-gravity occurred, a weight penalty of 760 lbs exists relative to the existing installation.

In the adaptation of the GSD-10 to the AAV-7A1, the improvement in steering performance, use of commercially available components, and increased system serviceability have to be weighed against the 760 lb weight penalty. Other configurations, such as a rear-drive scenario, will result in a more desirable system weight.

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1.0 SUMMARY

SEACO, Incorporated, under contract N00167-87-M-6177 with the David Taylor Naval Ship Research and Development Center (DTNSRDC) has completed an Engineering Design Study (EDS) for the adaptation, assembly, and installation of a regenerative mechanical differential steer drive system based on the Gleasman Steer Drive (GSD-10) for the AAV-7A1 tracked amphibious vehicle. Analyses conducted included (a) the collection and screening of applicable steer drive, transmission, brake, and control components, (b) development of a prototype steer unit configuration, (c) steer unit, engine, and transmission layout, (d) a summary of vehicle modifications, (e) weight and CG estimates, and (f) system performance studies.

The proposed steer drive and power transmission system has the following characteristics: (a) use of commercial transmissions and differentials, (b) positive steering control at all vehicle speeds, (c) separation of the steer drive and transmission unit, (d) a CG advantage but weight penalty associated with the new powertrain layout, (e) external (to the steer unit) brake system for air cooling and elimination of steer drive oil contamination, and (f) elimination of separate gearing for pivot steer.

A number of geometric arrangements were investigated. Engine compartment geometric constraints eliminated a number of engine/steer-drive/transmission layouts. Specifically, the longitudinal dimension of the engine compartment dictated a parallel engine and transmission layout, with the steer drive following (in-line) the transmission. Each major subsystem (engine, transmission, steer drive) can be removed from the vehicle independently. Although the present adaptation of the GSD-10 steer drive into the AAV-7A1 enhances steering performance, allows commercial components to be employed in the vehicle, and shifts the C.G. aft, the 1150 pound weight penalty of this distributed system is of concern.

2.0 INTRODUCTION

The importance of amphibious vehicles in military missions provides an impetus for a continued effort to design drive train components of light weight, high reliability, and high efficiency that are simple in design and versatile in application. The present AAV-7A1 amphibious vehicle may benefit substantially from a new driveline. The present work describes an Engineering Design Study (EDS) effort to adapt the Gleasman Steer Drive (GSD-10) vehicle steering system to the AAV-7A1. The GSD-10 provides true-pivot, regenerative steering with a mechanical system concept that is composed primarily of commercially available components. The use of off-the-shelf components can reduce development and tooling costs, significantly reduce final product cost, improve the availability of spares, and simplify maintenance.

Since the GSD-10 is a fixed gear reduction ratio steering system, the required sprocket speed-torque characteristics are achieved with the addition of a gear transmission to the system. In the present application the GSD-10 is the interface between a commercially-available, multi-speed, atuomatic-shift transmission and the AAV-7A1 final drive units.

2.1 Objective

The primary objective of the EDS was to identify the components and configuration of the GSD-10 steer unit and power train that would be of minimum weight, employ the space in the engine compartment in the most economical way, and have the least impact on the location and requirements of the auxiliary and support systems. It was clear that hull modifications were not desirable unless superior performance could be demonstrated, and that options which invaded troop space would decrease the utility of the vehicle.

2.2 Approach

The steer drive design and vehicle integration effort was performed as outlined below:

1. Select components whose performance would meet or exceed the requirements of the existing HS-400 system.
2. Eliminate those components whose size was not compatible with the existing AAV-7A1 engine compartment "real estate".
3. Establish an integrated system of minimum weight; determine impact on weight and C.G..
4. Base a control system on the optimal integrated system.
5. Establish retrofit requirements for integrating the new system into the vehicle.
6. Establish the new system performance expected.
7. Determine if any component corrosion problems were introduced.

System design and integration efforts followed these steps through two iterations; results for the first phase were presented at the 24 November 1986 NSRDC interim design review, the results from the second and final iteration are given in the present work.

2.3 SCOPE OF WORK

As set forth in the DTNSRDC Statement of Work, SEACO, Incorporated was contracted to accomplish the first phase of work under a three phase program as abstracted below:

Phase 1 - To adapt, optimize and design a mechanical steer-drive differential system for an AAV-7A1 vehicle installation against the following performance specifications:

- Vehicle weight: 2´ tons
- Maximum required level speed (land): 45 mph
- Capable of climbing a 60 % slope in low gear at 5 mph
- Acceleration of 0-20 mph in eight (8) seconds or less
- Vehicle engine - Cummins VT903T (400 bhp at 2800 rpm)
- Turn rate capability - 360 degrees in 8 seconds

The vehicle performance was specified to meet or exceed the current capabilities of the AAV-7A1 vehicle.

Phase 2 - To fabricate, acquire, and install the as designed Steer-Drive differential hardware and control components into an AAV-7A1 vehicle.

Phase 3 - To instrument, test, acquire, analyze, and report data pertaining to the Steer-Drive Differential and limited vehicle testing that will be performed at the Contractor's facility.

The present report covers the results of work performed under Phase 1. Within this phase, the following tasks were contained within the DTNSRDC contract:

- Identify and provide an overall description of the steer drive differential and transmission system. This effort includes key parameters such as operating speeds, power flow paths, proposed control method, etc. The drivetrain shall be for a fully amphibious vehicle and provide means of providing power to waterjets in the vehicle. Appropriate braking means and hardware for the vehicle shall be included in the design.

- Detail and describe the engine installation and modifications for mounting, cooling, controlling etc. At the culmination of the EDS, an installation design shall be prepared to show the proposed Steer-Drive Differential system placement in the vehicle giving rationale for component placement.

- The methodology, means and required hardware for controlling the steering differential and transmission shall be specified. This shall entail both mechanical and electrical components required.

- Vehicle performance shall be predicted on the basis of component performance and data.

- The control logic shall be illustrated in an analysis that gives the overall control envelope and defines critical response times and limitations. The Contractor shall provide for the engine operation to be controlled so as to provide optimum operation, fuel efficiency, and engine power requirements.

- A detailed weight estimate and center-of-gravity location estimate will be performed utilizing a Work Breakdown Structure (WBS) or similar tracking plan for components. This shall be started during the EDS phase and shall continue and be updated during the Option I - Fabrication and Testing.

- An estimate of transmission efficiencies in the vehicle's operating envelope in all three modes of vehicle operation (Land, Sea, and Land/Sea) will be made.

- Identification of major components (costing greater than $500.00), vendors, price, and availability/lead times.

- Component corrosion shall be addressed and preventative measures shall be taken to minimize this to the vehicle and components.

- The Contractor shall give consideration to the following areas when designing and integrating the Steer-Drive Differential and the AAV-7A1:

  - Human Factors Engineering
  - System Integration

- The subsystems and components that make up the system are not required to meet applicable US military specifications and standards since this is a research and development program.

Early technical results led to a series of meetings with Navy and Marine Corps personnel. Although the contractual scope of work was not altered in these meetings, the emphasis and direction was revised. The present report is a summary of Phase 1 design work conducted on the AAV-7A1.

3.0 STEER DRIVE AND POWER TRANSMISSION SYSTEM

Adaptation of the GSD-10 concept to the AAV-7A1 vehicle was performed with the approach outlined in Section 2.

3.1 Overview of Design Criteria

Table 1 contains a brief summary of the performance and design criteria for the AAV-7A1. Table 2 summarizes key performance criteria used in the adaptation of the GSD-10 to the AAV-7A1 application. In many areas, the GSD-10 is designed to improve performance relative to the existing HS-400 transmission. These improvements make the GSD-10 competitive with modern hydrostatic and hydrokinetic military transmissions.

3.1.1 Required Speed-Torque Envelope of the AAV-7A1 Vehicle: Land Mode. Figure 1 illustrates the breadth of the overall speed-torque envelope of the AAV-7A1 vehicle when it is operating in the land mode. The upper torque limit is set by the vehicle's tractive effort to weight ratio; a value of 1.2 has been set as a design criteria for the GSD-10 application since this was the value specified for the AAV-7A1. Under this extreme condition, the torque may be assumed to be equally divided between the two tracks. Thus, the tractive effort to weight ratio at each track is 0.6 for the case of rectilinear (straight-line) motion.

The maximum speed of each sprocket is determined by the highest vehicle speed, the track slip, and the desired turn radius at that speed. Since the AAV-7A1 vehicle is designed for a 45 mph top speed, the new powertrain employing the GSD-10 is designed to meet the same criteria. As a practical limit it is assumed the peak 45 mph vehicle speed cannot be maintained while the vehicle is turning; so therefore, the maximum sprocket speed need not exceed the peak rectilinear value. Given that the pitch radius of each AAV-7A1 drive sprockets is 10.5 inches (0.875 ft) and that the track experiences a 5% slip at the peak design speed the maximum sprocket angular velocity is 79.2 rad/sec. The abscissa on the plot shown in Figure 1 reflects the use of this value.

The corner horsepower is defined as the horsepower that would be required if the vehicle was capable of providing its maximum torque at its rated maximum speed. In the present vehicle the corner horsepower exceeds 3600 hp - this is an order of magnitude larger than the installed 400 horsepower of the present prime mover. After power to the fan, hydraulic pump, and alternator are subtracted (from the installed hp), the net propulsive power available is 165 hp/track in rectilinear motion. The large difference between net and corner horsepower is indicative of a requirement for a large "torque split" in the power transmission system. Here, the torque split is defined as the ratio of transmission torque capability for the powertrain in the vehicle transmission's lowest and highest gear. For the existing AAV-7A1 this means a split of approximately 22. The GSD-10 design is based on a similar split (but uses different transmission and torque converter ratios) and meets the maximum speed and torque conditions shown on Figure 1.

Figure 1 contains three lines of constant horsepower. The curve labeled 360 hp represents the maximum output of the engine after power to the cooling fan and alternator have been accounted for. The 165 hp curve represents the maximum horsepower delivered to each track in rectilinear motion (when driveline losses of 30 hp are assumed). In certain regenerative steer cases all 360 hp from the engine can be delivered to a single track while the opposite track regenerates 180 hp - this situation is described by the curve labeled 540 hp. Since the last case is considered the most severe sprocket output power, it is used to size differential components for a "skid-out" turn scenario at vehicle speeds approaching 45 mph.

For the purpose of defining required transmission reduction ratios, the 165 hp curve is useful. Shown in Figure 2 are speed-torque characteristics associated with transmission gear ratios of a typical automatic 5-speed transmission. The difference between the 165 HP curve and the delivered speed/torque curve is the energy loss in the driveline. A key influence on the shape of the commercial transmission curve is the presence of a torque converter in series with a multi-ratio automatic transmission. As shown in Figure 2, the torque converter switches in and out of its lockup mode as the vehicle accelerates through the transmission ratios. It is important that the curves have sufficient overlap to prevent the engine speed (linked to power output) from dropping a significant amount as the transmission shifts. This type of analysis leads to the conclusion that a minimum of 4 gear ratios are necessary to cover the broad speed-torque envelope of the AAV-7A1. Specific advantages and disadvantages of transmissions with more than 4 gear ratios are functions of variables such as shift time, steering characteristics, and engine torque curves.

3.1.2 Resistance to AAV-7A1 Motion. The parasitic force that consistently retards vehicle motion is the rolling resistance - the drag on the vehicle due to soil, track, and running gear losses. Rolling resistance varies with soil type, track tension, and running gear condition. In general, the rolling resistance varies from 80 to 130 lbs/ton of vehicle weight. For the purposes of sizing the GSD-10 for the AAV-7A1 it has been assumed the rolling resistance is 100 lbs/ton (2,400 lbs).

The energy available for vehicle acceleration is also affected by inertial effects, work to change elevation, and air resistance. Inertial loading (linear and rotational) occurs during the acceleration of the vehicle and its drivetrain. Once the vehicle has reached a steady-state speed, the kinetic energy (linear and rotational) can be computed. As noted in Table 1 the rotational energy can be approximated as an added mass term in the linear kinetic energy term. Hill ascending or descending forces can be computed from the work required to move from one elevation to another; a useful approximation is given by computing the change in the vehicle's potential energy.

Air resistance is ignored in many calculations since its typical contribution is roughly an order of magnitude less than the rolling resistance.

References 3 and 4 provide more detailed descriptions of the vehicle retarding forces and effects described above.

3.1.3 Steering Requirements. The over-land steering performance of a tracked vehicle can be described by the five regions shown in Figure 3 - the GSD-10 has been designed to operate in all five regions. Transition between regions is accomplished with the use of the vehicle steering wheel. For example, there is no need to shift the transmission to enter the counter-rotation steering mode (as is presently required on the AAV-7A1). All modes shown can be accessed regardless of the direction of vehicle motion (forward or reverse).

In military tracked vehicles the regenerative steering mode is of particular interest and importance. Since military tracked vehicles are designed to achieve and hold high speeds during maneuvers, the existance and efficiency of regenerative steering is of great importance. This mode is characterized by the transfer of regenerated power from the inboard track to the outboard track. In this situation the torque at the inboard track is negative (that is, must be absorbed or dissipated by the drivetrain) although its velocity remains positive. A major strength of the GSD-10 is the efficiency with which regenerative steering is accomplished.

As illustrated in Figure 4, vehicular rolling resistance is an important factor to consider in the analysis of high-speed turns. Rolling resistance on the inboard track provides a "drag" which is seen as a negative torque by the inboard drive sprocket. High-performance tracked vehicles have the capability to make tight turns at high speeds. Such turns require substantial negative torque at the inboard sprocket. Rather than waste the energy that would be dissipated in a mechanical brake, the GSD-10 regenerative system allows a large fraction of this energy to be captured and passed along to the outboard track.

Differential torque and differential power requirements to steer a vehicle vary with soil conditions; Figure 5 gives approximations for these parameters. In heavy sand and sharp turns the effective rolling resistance may reach 400 to 500 lbs/ton of vehicle weight (i.e., more than 4000 lbs of drag on each track). The increase in rolling resistance is associated with the binding of the track as a lateral force is devolped in a turn. For this reason, engine output power must increase to maintain speed in a turn even if the steering unit is 100% efficient.

Equations that describe turning performance yield solutions whose arguments include rolling resistance, vehicle mass and geometric characteristics, grade, and vehicle speed. To assure that a new driveline will meet the performance expectations, it is necessary to define a number of performance limits. In the design of the GSD-10 the key performance limits are summarized in Table 2.

Although the fundamental operation of the GSD-10 steer unit will be described in more detail in Section 3.2.1, the basic arrangement of the AAV-7A1/GSD-10 is shown in Figure 6. As shown in this Figure, the hydraulic motor provides the steering bias for the mechanical differentials. The maximum power of the motor is set by the amount of regenerative power that is transferred from the inboard track to the outboard track during a high-speed sharp turn. A review of performance data for modern transmissions indicated that the regenerative power level for a vehicle traveling at 45 mph in a sharp turn can reach 180 hp.

The maximum speed of the steer motor is set either by the need for rapid counter-rotation of the tracks at zero forward speed or by a sharp turn at relatively low speeds. Table 3 contains the definitions and computational procedure required to estimate the inner and outer sprocket speeds as a function of turn radius. The objective of these calculations is to predict the differential sprocket speed when the vehicle begins to "skid" laterally (in a sharp turn) as a function of the vehicle's forward velocity. As summarized in Table 4, the differential speed needed to skid the vehicle at low speed is very large. In the present powertrain design, the skid limit was set at 20 mph with a corresponding turn radius of 38 feet. Although definitive data was not obtained, this limit is thought to be fully competitive with modern hydrokinetic and hydromechanical transmissions.

The maximum torque of the steer motor is determined by the desired tractive effort to weight ratio that each track must generate with tracks counter-rotating. As discussed in later sections, the present GSD-10 has been designed to have a tractive effort to weight ratio of 0.5 per track for the case of counter-rotating tracks.

3.2 Description of the Proposed Steer Drive Unit and Power Transmission System

This section establishes component and interface options to provide the power transmission characteristics necessary to meet the performance goals of the AAV-7A1.

3.2.1 Overview of the GSD-10 Concept. The GSD-10 steer drive concept is based on a unique integration of two conventional gear differentials. A schematic highlighting the key system components and the parallel arrangement of the differentials is shown in Figure 7. A detailed mathematical analysis of this system is given in Reference 2, while an overview of some key system features is contained in this design integration report.

Note from Figure 7 that the GSD-10 offers the powertrain a fixed gear reduction ratio. In order for the power-train to provide the necessary vehicle speed-torque envelope, a gear transmission must be introduced into the powertrain; in the existing AAV-7A1 the transmission and steer unit are integrated in one unit. The GSD-10 design philosophy requires the identification of commercially available components, of which the transmission is the largest integral product. Desired transmission characteristics are described in Reference 5. The design integration effort to upgrade the AAV-7A1 with a GSD-10 steer drive system calls for (a) identification of the appropriate steer drive components, (b) integration of the components, and (c) introduction of a new control system; the present work is directed at accomplishing these three tasks.

Specific requirements which impact steer unit design are:

(a) continuous and controllable differential track speed,
(b) whatever power transmission capability other components in the system are capable of sustaining, and
(c) true-pivot, regenerative steering.

An expanded view of the GSD-10 system is shown in Figure 8. The conventional gear differential is a primary component in the GSD-10 system. A simplified view of a differential is shown in Figure 9. Two important features of a differential are shown in Figure 10. The kinematics of a condition where one wheel is fixed leads to the observation (see Reference 2) that a conventional differential cannot tell the difference between a vehicle losing traction and a vehicle entering a turn. The kinematics of a fixed driveshaft suggest the differential can provide true-pivot turning. The GSD-10 concept is based on the idea that, with the appropriate feedback control, a conventional gear differential is the ideal component on which to base a steer-drive design. Steer drive kinematics for rectilinear and curvilinear motion are shown in Figures 11 and 12 respectively.

Mechanically, the primary differential (power differential) distributes power from the transmission to the final drive units. Rigidly attached to the final drive shafts are a set of spur gears. These gears mesh with an identical set mounted on the secondary differential (steering differential). An idler gear is required for one gear train so that the system is kinematically correct. The output shafts of the primary and secondary differential lie parallel to each other. Auxiliary motor power passes through a reduction gear system to the steering differential and controls the relative velocity between the inboard and outboard secondary spur gears. In short, the steer motor is responsible for providing the motion of one track relative to the other and therefore the steering of the vehicle. It is extremely important to know the required steer motor power since the system is only viable when this load falls within the capabilities of the tracked vehicle prime mover. Steer motor power requirements represent the only power input (for steering) in the GSD-10 system.

3.2.2 GSD-10 Adaptation to the AAV-7A1. In the adaptation of the GSD-10 steer drive concept to the AAV-7A1 it is necessary to identify a new gear transmission and brake system for the vehicle. Although each subsystem (including the steer drive) must be designed for minimum size and weight for a given power transmission capability, each must also function as part of an integrated system. As such, the task of integration is an iterative one.

For the purpose of minimizing of AAV-7A1 vehicle modifications, a constraint was imposed on the adaptation of the GSD-10 that the powertrain be contained within the existing vehicle engine compartment. This precluded, for instance, a rear sprocket-drive concept. Such in-line concepts were of interest since a critical design problem was one to determine the component configuration which, for a given powertrain reconfiguration, would not create a bow-down C.G. shift.

Four basic powertrain configuration options considered in this work are shown in Figure 13. From the standpoint of weight, the configuration with the fewest couplings and transfer cases would be the first choice. Option 3 represents one such in-line configuration. However, in-line configurations do not fit within the existing engine enclosure. Options 1, 2, and 4 explore configurations that do fit within existing geometric constraints. Of these, 4 has the least weight.

The first design iteration (Figure 13, Option 1) attempted to minimize disturbances to the engine compartment by keeping the Cummins VT-400 engine and support systems in their original locations. Since this resulted in a non-optimal steer-drive integration, a significant weight penalty occurred (see Section 3.5). By the forth design iteration (Figure 13, Option 4) it became clear that if the engine was to be rotated 180 degrees and shifted over a few inches, then the transmission could be placed next to the engine. This allowed the transmission to output to interface directly with the steer unit that could now be placed in-line with the final drive units. Figure 14 illustrates this layout (details to be discussed in the sections to follow).

Specific components used in the GSD-10/AAV-7A1 adaptation were established by filtering candidates on the basis of power rating, size, and weight . In the event that inter-component integration problems existed (for instance, matching speed and torque limitations) alternate components were considered. A description of the results of this process for each subsystem follows.

3.2.3 <u>Steer</u> <u>Drive</u>. Some of the early recommendations for the GSD-10 configuration identified the steer drive as a ""bolt-together"" unit consisting primarily of commercially available differentials and gear transfer cases. An illustration of this approach is shown in Figure 15. This very early approach, although quite simple to assemble, proved to be heavier than could be allowed for a 400+ HP unit (preliminary weight estimates were for units of 300 HP). Upon further examination of various configurations and components, it became obvious that two factors worked against a low weight bolt together approach:

1. A penalty for redundant couplings, bearings, shafts, and structural casings was associated with the "bolt-together" assembly of off-the-shelf components, and
2. Component cases were a significant weight fraction of the differentials and tended not to employ light weight high-strength aluminium.

A design problem associated with a "distributed" driveline concept is the weight gain due to unavoidable redundancies. For example, the transmission is now divorced from the steer drive in the GSD-10 approach, and the sum weight of the two will inherently be heavier than that of an integrated unit for the reasons cited above. Structurally, the "bolt-together" approach also requires a chassis and will be more sensitive to alignment errors during installation.

In light of these drawbacks the appropriate step toward reducing steer-unit size, weight, and installation sensitivities was to consider commercially off-the-shelf differential cores and allow the cores to be integrated in a much lighter aluminium casing. This is referred to hereafter as the "integrated steer unit" approach. Later on, remarks about further weight reduction for a production unit reflect optimization of material in a cast unit.

Although potential steer drive components were screened initially by power capability, size, then weight, it quickly became clear the last two criteria could be performed simultaneously for the integrated steer unit approach. A significant difference (over the original concept) is increased flexibility in the configuration (internally) of the steer-drive components. One important modification is shown in Figure 14, where the ring gear on the primary differential has been reversed to permit the input pinion to lie more to the starboard side of the steer unit. This allows the transmission unit to be closer to the starboard side of the engine compartment so more room is available for the Cummins engine and its support systems.

Differential Selection. Candidate differentials for the steer drive are shown in Table 5. The important features of each are the reduction ratio and the power output per shaft. Selection of a gear differential is based on the unit's capability to accommodate the vehicle's maximum speed and torque requirements. These limits are established under the following loading conditions:

1. Maxumum engine speed and torque of, respectively, 2800 RPM and 825 ft-lb,
2. Maximum and minimum transmission ratios of 7.973 and 1.0, repectively,
3. Final drive reduction ratio of 3.06,
4. Peak sprocket speed of 720 RPM,
5. Peak sprocket torque based on a 1.2 TE/W ratio, and
6. Rectilinear motion.

The relatively simple procedure to compute differential input and output speeds and torques was easily mapped on a Lotus spreadsheet program. From an interface standpoint the "ideal" differential would not require any step-up or down of its input shaft since this would complicate the powertrain layout and add weight to the system. Interface requirements for the ideal differential are shown in Table 6a.

It is necessary in the interpretation of differential literature to emphasize the "per-shaft" differential output rating as opposed to the "total" output, and also that a differential not require a planetary reduction at the hub to achieve the rated output. With this in mind the array of suitable differentials drops off considerably. Many differentials which had adequate per-shaft output torque capabilities would, on the other hand, suffer unacceptably high input speeds to meet the maximum vehicle speed criterion. For the purpose of a balanced view of this affair the component interface spreadsheet (Table 6a) displays torque and speed data side-by-side.

Differentials selected for potential use can be compared by the gear reduction ratio they offer (all met or exceeded power requirements). Recalling the classic differential configuration, the gear ratio of the differential is precisely that of the pinion-ring gear reduction ratio. The differential reduction ratios for the present work range between 1.28 and 5.43. Based on preliminary information obtained from Leyland Gear Works, its unit with the 1.28 reduction ratio would allow a direct steer-unit/transmission interface (no step-up or down) and meet top vehicle speed requirements (see Table 6b). If this reduction ratio is passed over to the torque analysis the unit is required to handle a peak 5800 ft-lb ouput torque; verbal information obtained on the unit indicates this loading condition is satisfactory. Although the Dana-Spicer and Rockwell units were designed for vehicles of large axle weight and top vehicle speeds in the 55-75 MPH range, the major drawback to their use in the AAV-7A1 is the presence of the 3.06:1 final drive units. Specifically, application of the "as-built" differential would result in a (degraded) top vehicle speed of (approximately) 25 MPH, or conversely, the differential input speed would have to be stepped up by a factor of 4; in the latter case there exists serious thermal, lubrication, and component life effects and compromises. Since the bearings used in, for instance, the Dana-Spicer differentials have a dynamic load rating of 3000 hours at 500 RPM the speed factor associated with a 12000 RPM input speed degrades the life by almost 62% (under a 'normal' 2400 rpm input the life is 1875 hours for the 'standard' loading).

Distinct disadvantages associated with high (>3) differential reduction ratios prompted the idea to modify the ring-pinion set in, for example, the Dana-Spicer unit. Relevant to this modification is the understanding that differentials are normally rated on the basis of the bending stress for the ring-pinion gears; that is, the ring gears are designed to be the weakest link in the system. This avoids damage to the differential cage gear set. As such, the limitation to decreasing the the reduction ratio (and therefore the tooth stresses) is capped by rating the torque limit of the output shaft of the as-built differential. Estimates for a 1.5 inch diameter output shaft yield a section modulus per unit length of approximately 0.663 square inches, and an allowable torsional shearing stress between 67.5 and 150 thousand psi: the maximum allowable static twisting moment therefore lies between 5200 and 8300 ft-lb. In light of the 10,000 ft-lb (total) torque limit given for the Dana-Spicer unit, a conservative design would limit ring gear torque loads to the same. It is not recommended by differential manufacturers to operate under maximum torque conditions for 'extended' lengths of time, which typically means about 3 or 4 thousand cycles.

Implicit in this discussion of differentials requirements is an understanding of conditions under which the primary differential is expected to operate. In the GSD-10 it is possible for all of the prime mover power to pass through the secondary differential. It is also desirable that the primary and secondary differentials be interchangeable. This simplifies the logistics of spare part supplies and provides a conservative design.

Transfer Cases. What were previously identified as transfer cases in the steer drive now become simply a set of gear trains in the integrated approach. Quite clearly the services offered are the same. The transfer bearings, case, and lubrication system are integrated with those of the differentials. Gear pitch diameters of 6.125 were selected on the basis of the smallest allowable distance between the primary and secondary differentials. Shaft diameters of 1.5" are consistent with the estimated differential output shaft diameters.

Stee. -Drive Housing. For the prototype unit a welded housing made of 0.5" high-strength aluminum was to be assembled and internally partitioned as necessary for internal bearing supports and lubrication. Aluminum plate meeting ASTM specifications 5456 or 5083 will be used. Although it was desirable from a weight standpoint to minimize the housing wall thickness, it was also important to ensure the unit would not yield under severe loading conditions. From classical plate bending theory, a case of a 16 inch by 12 inch simply supported plate was studied, subject to a 3,000 lb load over a circular area 3 inches in diameter and located at the center of the plate. A deflection of 0.1 inch was computed and believed to be within the allowable limits of the housing structural requirements. Since the sides of the housing are probably cantilevered more than they are simply supported, and the bearing supports are not at the geometic center of the plate, the actual deflection is estimated somewhat lower than "0.1". In the event closer tolerances must be held, localized rib supports can be added to the housing structure.

3.2.4 Transmission. Selection of a new gear transmission unit for the AAV-7A1 was performed prior to focusing on the appropriate gear differential for the steer unit. The high and low-gear ratios of the transmission dictate the differential reduction ratio (or transmission speed step-up) required for the vehicle to meet the top speed criterion. Customizing a differential is much easier than modification of the transmission gear system and thus, transmission selection preceded differential selection. Important performance criteria for the transmission are its overall gear split, stall torque ratio, and input power rating; between any two units of comparable performance, weight and size would, of course, be deciding factors.

Among the transmission data surveyed, units which are off-the-shelf and currently in production were in the spirit of the present work (off-the-shelf designs have larger lead times). As such, the Detroit Diesel Allison DR750-series and the Renk Doromat 870-series transmissions were the best candidates for the GSD-10 adaptation effort. A summary of the key features of these units is shown in Table 7; detailed product information is provided in Appendix B.

Initially the design goal to minimize retrofit repercussions solicited an approach where a new gear transmission would interface the existing Twin-Disk Model 90192 torque converter. As the system design evolved it became clear that (a) reconfiguration of the prime mover would affect PTO and hydraulic pump interfaces anyway, (b) the Twin-Disk unit was not considered off-the-shelf, and (c) existing off-the-shelf automatic transmissions already have the torque converter as an integral part of the unit and are not easily mated to different torque converters. In summary, the existing Twin-Disk unit could not be satisfactorily integrated with the proposed transmission systems and was therefore eliminated.

The 2.83 stall torque ratio of the Twin-Disk unit is comparable to the torque converters which are part of the DDA transmissions. There is a variation between 1.83 and 3.04, depending on the DDA torque converter used, where the specific value is set from DDA's match between the type (and year) of the prime mover and the transmission of interest.

The RENK unit has a lower input power rating than the DDA unit. It can be argued that after accounting for all parasitic engine loads, the input to the transmission is not likely to see the rated 400 HP of the Cummins engine. Based on this the DDA unit is clearly overdesigned for the present application, while the RENK would need to be driven slightly above its design point. Since a conservative design would result from the use of the DDA 750 transmission, it was selected for the present work. Considerable weight savings do, however, make the RENK an attractive alternative, if a slightly lower tractive effort-to-weight ratio is acceptable.

3.2.5 Analysis of Brake Requirements.

As discussed earlier, the GSD-10 concept requires that a new brake system for the vehicle be identified - this provides an opportunity to improve the reliability of the vehicle by recognizing some shortcomings of the existing system. These primarily consist of brake oil filtration problems and the hinderance of servicing internally-mounted brakes.

The proposed system will employ air cooled disk brakes with hydraulically actuated calipers; some features of the system are:

(a) Elimination of the need for a cooling system which is shared by the steer unit,
(b) Elimination of oil filtering problems with the use of an air cooled system; any disintegration of pads or components is external to the steer drive mechanism,
(c) Use of commercial brakes that are low in cost and readily available,
(d) Bronze disks which are not prone to corrosion in a seawater environment.

Hydraulic acuation of the brakes is simplified by the presence of a hydraulic pump on the existing vehicle for auxiliary functions. The design of a control system is straightforward since it is a standard subsystem for a variety of on and off highway vehicles. The M-113 family of vehicles use such brakes allbeit at a lower gross vehicle weight.

Part of the motivation for a disk brake system were the geometric constraints in the AAV-7A1 engine compartment. A liquid-cooled brake system for the AAV-7A1 would require 6 to 8 inches of shaft length and be approximately 12 inches in diameter. The critical dimension here is the consumed shaft length, not the device diameter. A disk brake has the advantage of being thin, though of a larger diameter than the liquid-cooled system. Given the restriction for brakes to be confined in a cylindrical space of 17" radius and 3" length, air-cooled disk brakes were the rational alternative for the AAV-7A1.

Several approaches to sizing and performance estimates for disk brakes exist, but a clean summary of the essential steps is given by the <u>Twinflex Disk Brake Selection</u> notes in Appendix F. Consider the case of stopping the vehicle once per minute from an initial speed of 10 MPH; calculations conducted in accordance with Appendix F yield the following:

1. A braking torque of 87,000 in-lb per caliper shaft is based on:

(a) 48,031 lb vehicle weight,
(b) 14.67 ft/sec vehicle speed,
(c) 10.5 in sprocket radius,
(d) 3.06:1 final drive ratio, and
(e) terrain-track coefficient of friction of 0.7.

2. Mean power dissapation per stop of 650.4 HP.
3. Torque and thermal power dissipation suggest a Twinflex Model VC caliper and a 24 inch diameter, 1.0 inch thick disk.

4. Characteristic temperatures:

(a) disk surface temperature rise per stop of $259^{\circ}F$, and
(b) a maximum disk temperature of approximately $1,000^{\circ}F$.

5. Approximate pad life of 648 stops (10.8 braking hours).

Since pad life is directly proportional to the number of disks the approximate pad life can be extended as desired. An upper limit to this is presented by the geometric constraints within the engine compartment.

3.2.6 AAV-7A1 Power Train Modifications. To accommodate the new steer drive system key AAV-7A1 powertrain elements must be reconfigured; a secondary effect is the redesign of support systems interfaces, e.g., fuel system lines, intake air ducts, water cooling, and exhaust systems. Modifications are addressed in chronological order of intallation:

(a) Transmission,
(b) Prime mover,
(c) Steer Unit,
(d) Brake system.

In light of the iterative nature of the EDS effort it was efficient and cost effective to develop a CAD data base for the vehicle and components that were under consideration. Many of the subtle intallation details associated with the GSD-10 adaptation to the AAV-7A1 can be found on the prints themselves; the three perspectives of the installed system are given in Appendix D as foldouts in the pouch in the back of this document. The descriptions below are intended only to highlight some of the most important features of the AAV-7A1 modifications that would take place.

Estimates on the characteristic time for installation have not been made, nor has a detailed design of the auxiliary system been performed.

Transmission. As shown in Figure 14, the DDA HT750DR is positioned slightly under the weapons station and is therefore recommended as the first component to install. As presently configured, the transmission will employ the existing DDA input cover, but a bearing set must be added to support the transverse loads associated with the new interface with the prime mover. Splined shafts through chain sprockets are mounted in the bearings to drive a Morse "Hi-Vo" chain. The bearings are of the roller-type. A separate lubrication system will be used for the chain drive, but the enclosure and bearing supports will be an integrated part of the transmission housing. Transmission support mounts will be attached to the same cross-members on the AAV-7A1 vehicle hull that the Cummins engine is mounted to.

Prime Mover. The prime mover installation will follow that of the transmission. It will be necessary to rotate the engine 180 degrees from the existing configuration and shift its support mounts 10 inches to the port side of the vehicle; the crankshaft pulley that one could ordinarily view from within the troop compartment will now face toward the bow.

Very important in the reconfiguration are the impacts on driver's and troop compartments. In order to make room for the engine in this new layout, the bulkhead in the passage leading to the driver's and troop commander's seats must be moved three inches port. Inspection of a typical AAV suggests this makes the passage somewhat tight, but does not restrict passage. Depending on the shift in position of the air intake and cooling system it may be possible to taper the 3 inch bulkhead shift so that the forwardmost portion of the driver's station is not dramatically affected. It was not established what design margins the existing passage width allowed with respect to MIL-H-46855, MIL-STD-1472, and MIL-HDBK-759.

A benefit of the proposed layout is that very economical use has been made of the longitudinal engine compartment space. This means the engine can be allowed to shift more to the bow were it not for C.G. considerations. Recognizing that the DDA750 series transmission is somewhat over sized (in power) for the present application, the proposed layout may be the direction to head in as higher horsepower concepts are considered.

In order to accommodate the new position of the engine, the cradle on which it is mounted must also be rotated and moved forward; it also must be extended to allow for the chain sprocket bearing supports to be added. Once assembled, the engine, transfer case, and transmission effectively become one structural unit.

With the reversal of the prime mover the cooling tower must be modified since the tubocharger now penetrates the area dedicated to the radiator. Two possible methods to correct this situation are (a) the turbocharger can be lowered as it is on the marine version of the Cummins engine, or (b) the cooling tower redesigned. The former would require the intake manifold to straddle the flywheel housing and the engine to therefore become a customized unit; the latter has the potential to become an extensive retrofit task.

Steer Drive. As currently envisioned, the steer drive would be installed last, and is located in the vicinity of the HS-400 unit which it replaces. This position allows the output of the GSD-10 to interface directly with the final drive input couplings. The steer drive input is received directly from the DDA transmission output; no step-up or down is required. Allowance is made in the mounting of the steer drive to permit the unit to accommodate anticipated vibration and compliance of the final drive units. The existing universal joints have been retained to minimize alignment problems. The hydraulic pump used to supply the steer motor is mounted on the hull as close to the center line of the engine crankshaft as possible. Since the forward bearing set of the engine is incapable of enduring significant radial loads and the hydraulic pump shaft is at a lower elevation than the engine crankshaft, a drop-box was used. It was convenient to locate the heat exchanger and hydraulic reservoir on top of the steer drive since it would then be in close proximity to the pump - this also minimized unnnecessary plumbing and lengths of high pressure hydraulic lines.

Brake System. Disk brake systems of the type to be employed in the present work are rather simple to install. Tenatively the disks will be splined on the output shafts outboard of the U-joint yokes, as shown in Appendix D. The calipers would be mounted on the hull and their respective sponsons. Mounts for the calipers will be welded to the hull.

Support System Modifications. As noted earlier, the GSD-10 installation requires major alterations in the engine compartment. The engine has been rotated 180 degrees and moved laterally, the turbocharger has been moved, engine and driveline foundations redesigned, and the firewall on the driver's side of the vehicle shifted over several inches. These alterations are necessitated by basic geometric differences between the GSD-10 and the existing HS-400 transmission. With the changes in the overall engine room layout have come associated changes in the location of the alternator, cooling fan drive, hydraulic pump, batteries, air cleaner, and the electrical, mechanical, hydraulic and air conduits that run to and from the associated support systems. Such alterations constitute a major redesign of the engine compartment. Proposed alterations in the support systems may be reviewed in the drawings of Appendix D. A detailed design ancilliary systems must be performed before an AAV-7A1 installation is attempted.

3.3 Steer Drive and Transmission Subsystem Control

The Gleasman Steer Drive has been conceived under the premise that a competitive tracked vehicle driveline can be assembled from commercially available components. This philosophy is carried into the control system where efforts have been made to adapt existing transmission controllers and hydraulic components to the present tracked vehicle application. The control of an amphibious tracked vehicle is somewhat more complex given the need to operate in three distinct modes.

In the land-mode of operation, the active components of the driveline operate as they would in any tracked military vehicle. The driveline must "condition" the engine output speed and torque to match the speed and torque required at each of the two track drive sprockets. The differential speed of the two drive sprockets must be accurately controlled such that the vehicle can steer within the five regions defined in Figure 3.

In the "water-mode" of operation, the tracks do not rotate. Rather, two waterjets are used to propel the vehicle as indicated in Figure 6. The two waterjets are relatively simple to control. As presently configured, the waterjets are driven through a power-take-off located between the engine and the torque converter of the HS-400 (FMC) transmission. The speed of both waterjets is proportional to the speed of the engine. The power absorbed at the waterjet is roughly proportional to the rotational speed of the jets raised to the 2.5 power. In-water steering is accomplished with steering buckets or deflectors. These buckets are deployed at the nozzle of each jet. As they are deployed, they deflect the jet discharge. When partially deployed, each bucket can direct the jet discharge athwartships. When fully deployed, each jet can be deflected through an angle of nearly 180 degrees such that the vehicle can obtain "reverse" thrust.

A third mode is required for transitioning from open water operation to operation ashore. This mode is known as the transition or surf mode of the driveline. In the surf mode, both the tracks and the waterjets must operate. Furthermore, both the steering buckets and the differential track speed control systems must be operational. It is crucial that the tracks be operable as the vehicle first makes contact with the beach in the surf. As contact is made with the tracks rotating, the vehicle is pulled forward and positive control over broaching is maintained. In the surf-mode of operation, relatively little power is required to rotate the tracks until contact is made with the beach. Once contact is made, the load shifts to the tracks. Control over the differential track speed and the steering buckets must be maintained in parallel in the transition mode of operation.

3.3.1 Vehicle Mode Control. The operator must be capable of selecting any one of the three operating modes (i.e. land, surf or water) from within the driver's control station. It is preferable that all mode changes be allowed with the vehicle in motion. For example, the vehicle should not be required to stop in the water when transitioning from water-mode to surf-mode, nor should it be required to stop once on firm land to change into the land-mode.

From a human factors standpoint, the selection of a new driveline mode must require deliberate action on the part of the operator. There are many hazards associated with inadvertent mode changes. Mechanisms such as detents, mode change interlocks, etc. are required to prevent inadvertent mode changes.

Land Mode Control. As shown in Figure 6, the GSD-10 driveline has both mechanical and hydraulic power paths. When operating in the land mode, with the vehicle experiencing rectilinear motion, the power path is entirely mechnaical. When operating in the land-mode in curvilinear motion, the primary differential continues to receive mechanical power while "steering" power is delivered to the secondary differential by a hydraulic motor.

The land mode is selected by moving the shift control lever into either "forward" or "reverse". A toggle switch is located on the operator's console. The three-position toggle switch is marked "land", "surf" and "water". When in the land position, valves are closed to block the flow of oil to the waterjet drive motors.

Water Mode Control. The water mode is selected by moving the mode control toggle switch to the "water" position. The position of the shift control lever is irrelevent since its functions are inoperable when the vehicle is in the water mode. Thus, any position of the shift control lever is allowed when the vehicle is shifted to the water mode.

In the water mode, valves are opened to allow the flow of oil to the waterjet drive motors. A valve in the piping to the land steer motor is closed to prevent it from providing differential track speed in the water mode. In the water mode, the clutches within the mechanical transmission are held open such that no power can flow through the mechanical power path. Power is taken from the engine by the hydraulic pump. Oil is split in a flow divider such that the two waterjet drive motors operate at approximately equal speeds. Steering control is maintained with the deflecting buckets at the discharge of the jets.

In the water mode, the variable displacement (full-over-center) pump must supply oil to the fixed-displacement waterjet drive motors. Since the waterjets are designed to rotate in only one direction, the displacement of the pump is allowed on only one side of "center". This feature is built into the analog circuit logic that is activated when the mode select toggle switch is moved to the water position. In effect, the control voltage to the pump displacement control servo valve is allowed to vary only from 0 to 24 volts as opposed to the -24 to +24 volt range associated with land steering.

In the water mode, the displacement of the hydraulic pump and the speed of the engine are proportional to the displacement of the conventional accelerator pedal beneath the operator's right foot.

Surf Mode. The surf mode requires more complex control. When the operator moves the mode control switch to the surf position, the land and water drivelines must operate in parallel. This implies that there are fluctuating demands for hydraulic power to the land steer motor in parallel with the need for hydraulic power to the waterjets. Such fluctuations in two parallel systems can cause resonance if both engine speed and pump displacement are free to react to the varying load. For this reason, the pump displacement feedback signal, in conjunction with engine operation at a constant throttle setting, can be used to assure control system stability when the vehicle is in the surf mode.

The vehicle operates in the surf mode for a small percentage of its operating life. In this mode, the need for track speed is limited to approximately 10 miles per hour. To avoid broaching in shallow water, the steering system must be able to respond rapidly to the operator's steering commands. The response must be equally effective with the deflector buckets or the differential track speed.

As shown in Figure 16, selection of the surf mode opens the supply/return valves to both the waterjet drive motors and to the steering motor. Unlike the land mode, the hydraulic pump can only displace in one direction when the waterjet drive motors are in the circuit. For this reason, the normal method of controlling land-steering direction and speed cannot be used. Rather, the pump is always displaced in one direction and pilot operated flow control valves must be used to port the oil to one side or the other of the steer motor. Flow control valves in the motor supply lines limit track speed and thus assure sufficient oil to the waterjet drive motors. Fortunately, the quantities of oil required for steering at the relatively low speed of 10 mph are small and can be handled within valves of reasonable size.

When the mode selection switch is put in the surf position, a number of hydraulic valves become active to establish the required flow paths and degree of steering control. The resulting control is less satisfactory than the land-mode control but is adequate to provide responsive steering control when the vehicle is in the surf zone.

3.3.2 Engine Speed/Power Control. The Cummins VT-400 engine is equipped with a governor that provides the operator with control of engine speed and power in a single pedal movement. The governor system will be used as it is presently configured. A feedback signal giving data on present throttle position will be provided to the transmission. With such a feedback, the decisions on up-shifting or down-shifting can be made by the transmission with knowledge of current governor position.

3.3.3 Transmission Shift Control. As proposed for our baseline concept, the transmission is an Allison HT 750 DR. This transmission has a rating of 450 hp, has five forward speeds, one reverse speed, an integral torque converter, and an optional hydraulic retarder. The transmission receives a throttle position signal from the engine. The shift sequence is then a function of the throttle position and the actual engine speed. The shift sequence is controlled within the transmission by a series of mechanical and hydraulic linkages. Allison also has electronic controllers for this transmission. For reasons discussed in the following paragraph, the electronic control is preferred.

As the vehicle steers, power is extracted from the engine by the hydraulic pump. Such power extraction reduces the power available to the transmission input. Such circumstances exist in commercial applications (e.g. concrete delivery and refuse trucks). However, the present application requires larger variations in ancilliary power demand at times when the transmission is also exerting a high power demand. Thus, some customization of the standard electronic control package will be required. The hydraulic pump will be provided with a displacement feedback signal. By monitoring pump displacement and pump discharge pressure, the power being drawn by the pump can be easily computed. This value can then be used to bias the upshift/downshift control package to account for the presence of the relatively large hydraulic pump.

3.3.4 Vehicle On-Land Steering Control. Control of the vehicle direction when operating on-land is accomplished by varying the displacement of the hydraulic pump. The pump speed is always proportional to engine speed. Pump speed is not an independent variable. If straight-line motion is desired, the pump remains at zero displacement and no oil is delivered to the fixed-displacement hydraulic steer motor. Consequently, the steer motor does not rotate.

When curvilinear motion is desired, the pump is stroked to deliver oil to the steer motor. The amount of oil pumped to the steer motor, and hence the turn radius, is a function of engine speed. In past DTNSRDC designs, the displacement of the steering wheel has been proportional to vehicle turn radius. In the present design, the amount of pump stroke is linearly proportional to steering wheel angular displacement. The operator is the source of feedback to compensate for changes in engine speed. There is no electronic correction to account for variations in turn radius with engine speed. With this approach, the control system avoids complex algorithms necessary to prevent resonant conditions involving variations in engine speed, steering wheel position, pump stroke and vehicle speed.

The hydraulic pump may be displaced in a positive or a negative direction. If displaced in a positive direction, oil is supplied to one side of the hydraulic steer motor. Pump displacement in the negative direction reverses the flow of oil, thereby reversing the direction of hydraulic motor rotation. Thus, the direction of pump displacement determines the direction of motor rotation and hence the direction of vehicle steering.

Pump displacement is physically controlled by an electrohydraulic servo valve mounted on the pump. The voltage supplied to the valve is linearly proportional to the displacement of the pump. An ABEX pump (14 cubic inches per revolution) was selected because of its superior control characteristics. The ABEX pump has two built-in but separate pumps within the main pump. One of these auxiliary pumps serves as a boost pressure pump to assure that the suction of the main pump remains pressurized at all times (i.e. a charge pump). The second pump is a dedicated source of control pressure for the hydraulic servo valve. Other manufacturers combine the two functions in a single pump. The result is a fluctuation in the control pressure to the servo valve whenever there is a rapid change in the requirement for charge pump oil. The ABEX pump is relatively insensitive to such changes and therefore exhibits more constant and linear control characteristics.

The ABEX pump also contains a feedback potentiometer that supplies a signal proportional to actual pump displacement. This signal allows the construction of an analog feedback between the desired pump displacement (i.e. the supplied voltage) and the actual pump displacement (i.e. as measured by the potentiometer). When combined with the separate control pressure pump, this feature provides a very accurate, linear and repeatable relationship between the operator's signal and the resulting pump displacement. This, in-turn, results in the most positive operator "feel" for control of the vehicle.

The proposed method of obtaining land-steering is reliable, rugged and is used in numerous commercial and military applications. The control valves and algorithms are simple and proven. The main failure modes are associated with faults that would prevent the hydraulic pump and/or motor from operating. In these failure modes, the vehicle would not steer at all.

Although such failures are not desirable, they are preferable to a failure mode in which the vehicle tends to steer erratically.

3.3.5 In-Water Steering Control. The AAV-7A1 has an auxiliary hydraulic pump which supplies power to raise/lower the ramp, actuate the plenums, operate two bilge pumps and control the in-water steering. This pump will be retained and its functions will remain unchanged. When a signal is provided by the operator to the water-steer-control valve, oil is routed to one of the actuators on the steering buckets and the bucket is deflected to the desired position. This system will be retained. The only change in the system will be the use of hydraulic motors to drive the waterjets.

3.3.6 Hydraulic Motor Selection. Selection of the steer motor is an initial step in the application of the GSD-10. Since hydraulic motors are rated by maximum speed, maximum torque and maximum continuous power, it is necessary to translate the vehicle performance requirements into performance required by the hydraulic motor.

Motor Performance Requirements. As described in Section 3.1.3, the maximum speed of the hydraulic motor was chosen to give the vehicle a capability to enter a skidding turn at speeds as low as 20 mph assuming a coefficient of friction of 0.7. Table 4 shows a turn radius of 38 feet and a differential track speed of 77.75 rpm at this condition. At this point in the design, the reduction ratio between the steer motor and the sprocket was not fixed. In later steps, the maximum speed of a candidate motor will be divided by the differential sprocket speed to obtain an overall reduction ratio.

With a tentative differential track speed, the yaw rate of the vehicle in a counter-rotating turn is determined as outlined in Table 9. With a differential track speed of 77.75 rpm, a yaw rate of 45 degrees per second was obtained. This yaw rate is thought to be an improvement over present AAV-7A1 performance.

Hydraulic Motor Candidates. A survey of available hydraulic motors was conducted. Manufacturer's literature was used to characterize the performance of each candidate motor. Motors were split into two categories: those with fixed displacement and those with variable displacement.

Fixed Displacement Motors. These motors are able to operate at higher speeds since internal components are rigidly mounted and are better able to withstand forces associated with high rotational speeds. Since displacement cannot be lessened as speed increases, these motors require larger pumps to attain high output speeds. In addition, the average hydraulic pressure tends to be lower in a fixed displacement motor. If the pressure falls below approximately 1500 psi, the mechanical losses become excessive. For this reason, fixed displacement motors may be somewhat less efficient in the present application. There are no control functions associated with a fixed displacement motor and thus the control of the pump/motor system is much simpler, more reliable and less costly.

<u>Variable</u> <u>Displacement</u> <u>Motors</u>. Earlier studies by DTNSRDC have revealed control complexities associated with simultaneous control of a variable displacement pump and motor. The variable displacement motor requires a smaller pump but is typically limited to lower output speeds. Since variable displacement motors have more moving components, they have lower reliability.

Tables 10 and 11 summarize the results of the hydraulic motor survey. All motors that offered high output power at low weight are of the axial piston design. A comparison of Tables 10 and 11 illustrates the higher output speeds of fixed displacement designs.

<u>Hydraulic</u> <u>Motor</u> <u>Selection</u>. The steps used in selecting a hydraulic motor are outlined below. The maximum motor speed is first used to establish the overall reduction ratio (i.e. hydraulic motor to sprocket) needed to meet the differential sprocket speed requirement. This ratio is then multiplied by the maximum motor output torque to obtain the maximum differential sprocket torque that the motor can generate. The overall weight, power and power to weight ratio of each motor were then computed. Results are summarized for variable displacement motors in Table 12 and for fixed displacement motors in Table 13.

Screening criteria were established to narrow the list of candidate motors. These criteria, as summarized in Table 1, included weight, power and torque limits. The torque limit corresponds to a tractive effort to weight ratio of 0.5 per track as set early in the project (See Table 2). Motors that meet the screening criteria are listed in Table 15.

From the candidates in Table 15, the most promising fixed displacement and variable displacement motors were selected. The Volvo 6.72 cubic inch motor was selected as the fixed displacement candidate while the Rexroth 9.76 cubic inch motor was selected as the variable displacement candidate. The Rexroth selection was driven weight, speed and by the high speed and light weight capability of the motor relative to other variable displacement candidates. The Volvo selection was driven weight, speed and by the presence of piston rings in the motor design. The rings yield a higher volumetric efficiency and allow a greater piston to cylinder clearance.

As summarized in Table 16, the fixed displacement motor was selected. The selection was made when it was recognized that the large hydraulic pump associated with the steer motor could also be employed to drive the waterjets in surf and sea modes of operation. This approach eliminates the need for a power takeoff on the transmission torque converter. Since commercial transmissions do not come equipped with PTOs of sufficient capacity, a custom design would have been required to support the water drive requirement.

Hydraulic Pump Selection. Selection of a pump was relatively simple after selection of the hydraulic motor. The sizing of the pump was driven by a desire to retain full steering performance at reduced engine speeds (i.e. down to 1800 rpm). A major criticism of the present transmission is the loss of steering performance as engine speed drops. For example, steering is effectively lost in a crash stop. For this reason, a hydraulic pump with approximately twice the displacement of the hydraulic motor was sought. The motor can then operate at full speed with the engine operating at 1450 rpm.

A second driving factor in the selection of the pump was the linearity of pump response to an electrical signal. The ABEX 14 cubic inch pump was ultimately selected for its unique features which enhance the linearity of pump response. These features include an electrical feedback signal proportional to actual pump displacement and a separate control pressure pump that is independent of the charge pressure pump. When coupled with a fixed displacement motor, such a pump offers simple, reliable and repeatable control characteristics.

3.3.7 Hydraulic Motor/Steer Drive Interface. The hydrostatic motor will drive into the steer drive unit through a high-ratio gear set. The present analysis provides an analysis of the size and weight of the required gear set.

The Volvo hydraulic motor has the following performance characteristics:

Maximum Input Speed: 3300 rpm
Minimum Input Speed: -3300 rpm
Maximum Input Torque: 534 lb-ft
Motor Displacement: 6.72 cu-in (fixed)
Intermittent Motor Output Power: 240 hp
Continuous Maximum Motor Output Power: 190 hp
Average Motor Output Power: 40 hp
Operating Hours per Year: 200
Operating Hours per Lifecycle: 2000 (estimated)

To provide the required differential output speed of 77.7 rpm at the vehicle sprockets, the interface gear must have an internal ratio between input and output of 13.88:1. This ratio, in series with the 3.06:1 final drive ratio provides the required overall ratio of 42.47:1.

The 13.88:1 reduction ratio between the hydraulic motor and the steer differential is accomplished in two stages. A spur gear set at the output of the motor provides a 4.63:1 reduction. The output of this gear set goes into a 3:1 ratio level gear set.

The hydraulic motor, acting through this two stage reduction, is effectively "self-locking" just as a worm gear would be. The use of the two-stage gear set offers a substantial efficiency improvement over the single-stage worm gear set.

In other applications of the Gleasman Steer Drive, where electric steering motors are used, it is critical that the interface gearing between the motor and the steer drive unit be "self-locking". In the present application, where a hydraulic motor is used, the motor can provide the locking feature.

3.3.8 Summary of Control System Components. In keeping with the "commercial-component" design philosophy, control systems have been selected for their availability, performance and compatibility with hydraulic and mechanical components. The following control items will be employed:

VT-400 Engine: Existing governer
Hydraulic Pump Displacement: ABEX Current-Loop, Electro-Hydraulic stroker with pump feedback (See Appendix B).
Transmission Controller: DDA Mechanical & Electronic, Custom features to accommodate large hydraulic pump, Inputs include engine speed and engine throttle position. Automatic upshift and downshift through all five ratios with option for manual select of first or second ratio. Manual selection of reverse ratio.
Vehicle Controls:
Directional Control: Existing steering wheel modified to provide current-loop electrical output signal.
Transmission Control: Existing gearshift lever with modified position markers. Position for counter rotation removed.
Speed Control: Existing accelerator pedal, no modifications.
Mode Control: Three position toggle switch with position locks. Hydraulic valves as per Figure 16.
Alarms: No modifications, add a low hydraulic oil and high hydraulic oil temperature alarm.

3.4 AAV-7A1 Performance Study

3.4.1 Overview. A vehicle performance study was conducted to estimate the vehicle's top speed, gradability, and dynamic response. As described in Section 3.2, top speed and gradability are integral to the selection of the gear differential unit used in the GSD-10; performance implications of the approximations made in that section are discussed below. The focus of the dynamic response analysis was on the determination of system shock loading and the effect of component inertia on vehicle acceleration.

3.4.2 Top Speed and Gradability.

Maximum Speed. From a kinematic point of view, the maximum vehicle speed can be given simply as the product of the engine speed, overall powertrain gear reduction ratio, and the pitch radius of the drive sprocket. This approach (though a good preliminary step) fails to account for a number of physical effects that significantly affect top speed capability. As a practical matter, vehicle speed is a maximum at the point where the net propulsive energy to the vehicle tracks is balanced by energy expended in overcoming physically retarding forces - in the present work the latter are primarily rolling resistance and aerodynamic drag.

As an example, consider a rolling resistance of 82 lb/ton and a power train transmissivity of 0.7 (power delivered to the sprocket is the product of 0.7 and engine BHP). A balance of energy yields:

$$(0.7\ BHP) - (W\ K_{RR}\ ) - (q\ C_d\ \ A)U = 0$$

where

BHP - brake horsepower output of prime mover
W - weight if vehicle
K_{RR} - rolling resistance coefficient
q - dynamic pressure
C_D - drag coefficient
A - projected frontal area of AAV-7A1
V - vehicle velocity.

Recognizing that the dynamic pressure is a function of the square of the vehicle velocity, top speed is solved from the cubic equation for velocity given above. From the solution to the polynomial (see Appendix C.1) yields a top speed of slightly more than 40 MPH. Computations are much easier by recognizing that a force balance is the kinetic equivalent of a (steady state) balance of power. As such, a simpler formulation for top speed is given by:

$$(M_{SP}/R\ \ ) - F_{RR}\ \ - F_{DRAG}\ \ = 0$$

where

M_{SP} - sprocket torque
F_{RR} - rolling resistance
F_{DRAG} - drag

This requires no rootfinding to find U and is easily mapped on a spreadsheet format; Table 20 illustrates the rolling resistance and powertrain gear reduction requirements associated with top vehicle speeds between 30 and 45 MPH.

In high gear (5th gear for the DDA transmission) a top vehicle speed of 45 MPH can be achieved if the rolling resistance is approximately 77 lb/ton. If it was desired to accommodate much higher rolling resistances, the sprocket torques would greatly exceed what the vehicle could presently offer (see Table 20).

Gradability. Under a no-slip track assumption and a coefficient of friction of 0.7 for the terrain-track interface, Table 21 illustrates the vehicle can entertain a grade of about 65% for a rolling resistance of 80 lb/ton and a grade of nearly 63% in the presence of a 120 lb/ton rolling resistance. The calculation procedure for gradability can be found, for instance, in References 3 or 4.

3.4.3 Dynamic Characteristics. As mentioned previously, the dynamic analysis focused primarily on the determination of system shock loads and on the dynamic response of the vehicle. For this limited purpose the Twente University of Technology Simulation (TUTSIM) was employed. Although used primarily for control system analysis, this digital simulation of analog circuits is extremely easy to implement on a PC-based computer system and found convenient in the analysis of the GSD-10 system.

Overview. The generic approximation to the components of the AAV-7A1 powertrain for the present work is shown in Figure 17. A block diagram for the complete system, including the control interfaces, is shown in Figure 18.

Two basic approaches exist for vehicle simulation efforts. In the first, a vehicle operational profile could be prescribed (a "test track" over which the vehicle would be run) and the required speeds and torques at the drive sprocket determined. The required powertrain capabilities would then be established, working backwards from the sprocket, through the final drive, the steer drive, and finally through the transmission to the engine. In the second approach, which was used in the present work, loading on the drive spocket is obtained by working forward from the engine, through the transmission, and so on to the sprocket. Although this approach asks difficult questions concerning engine throttle and acceleration characteristics, is was felt to be more amicable in establishing specific component speed and torque loadings.

TUTSIM Simulation Approach. The lamination of the general system block diagram (Figure 18) on the TUTSIM approach to system analysis is given for specific subsystems in Figures 19a - 19e. Of specific interest are the prime mover simulation (Figure 19a) and the transmission simulation (Figure 19b). In line with the goal of a simulation with the focus described previously, some simplifying assumptions have been made.

In the case of the prime mover an elementary engine acceleration profile was chosen; the approximation does not account for feedback loading from the remainder of the powertrain except for the speed drop associated with the transmission as it shifted up through the gear sets. As such, this assumes the engine was capable of providing the prescribed acceleration characteristics, independent of the effect of this on the remainder of the system. Real systems are quite different. To compensate, very high acceleration rates were chosen so there would be a conservative tendency to load the system more than it would otherwise. Curvefits to Cummins engine data were used for the simulation of the VT400 combustion characteristics.

Closely related to the lack of torque feedback to the engine was the simplifying assumtion made for the transmission torque converter. The torque ratio (quotient of the transmission torque and the engine torque) and the speed ratio (quotient of the turbine and engine speeds) relationship employed in the present simulation is shown in Figure 19b. Implementation of this approximation allowed turbine speed to be fedback to the pump and therefore dictate the torque ratio, but not the converse. As such, the transmission simulation cannot be used for the analysis of down-shifting.

Inertial loadings are an important element of the system in its transient state, and loadings for the transmission, steer drive, final drive and drive sprocket are easily accommodated; the implementation in the present work is shown in Modules P3 - P5 of Figure 19c. quite clearly. It is implied that the sprocket inertia contains that of the vehicle tracks. Literature from the manufacturers of the components considered for adaptation to the GSD-10 did not include the driveline inertias of interest, so the computer program in Appendix C.2 was written for this purpose. Tables 23, 24, and 25 provide approximate inertia values for selected components.

Two TUTSIM input files were generated - the first (AAV&100) fixes the transmission gear to a specified value, while the second allows automatic shifting in accordance with the schedule of Figure 19b. Appendix C.3 contains the input files and the numerical output from a typical run. Key input parameters for the simulation are given in Table 22. Central to the simulation theme of figure 18 is the sum of forces around the drive sprocket, as shown in Figure 19d. The environmental loadings for the completion of the sprocket force balance are described in more detail in Figure 19e.

Results. The four test cases considered provided an overview of the effect on rate of gear shifting, shock loads in the system, vehicle and engine speed, and the torque on the steer drive output shaft (total).

In the first case (Figure 20), the system behavior with no rolling resistance or inertial effects (except for the transmission) were investigated. As expected, the vehicle responded quite quickly to engine acceleration and shifted up to 5th gear in a relatively short period of time. Steer drive output loads approached 4,000 lb per shaft for a fraction of a second at the start of the simulation. It should be noted the simulation views the engagement of gears and associated torque application as a step function. As a consequence, rates of component acceleration is quite high and the oscillatory response (physical and numerical) is evident. This situation is somewhat like "popping" the clutch on a manual transmission which is not quite what occurs in the automatics.

Figure 21 illustrates maximum torque loadings are not as large when a rolling resistance of 80 lb/ton is accounted for, but there is a significant decrease in the response of the vehicle to engine acceleration. The steer drive output torque curve reflects minor shock loading at the points where the transmission gear changes and the engine speed remains constant. Significantly more oscillations and larger shock loads are evident when steer drive and final drive inertias are included, as shown in Figure 22. Instantaneous loading at the time of a gear shift is on the order of +/- 500 lbs, and is somewhat smaller than expected (given the severity of the approximations involved).

The last case considers a 1000 ft-lb-s2 effective sprocket inertia that incorporates some of the track rotational effects. This case, shown in Figure 23, yields highly oscillatory steer drive torque output loadings over most of the period of the run. Although some speed and torque phase oscillations are expected during gear shifting (see reference 6 for more details) continuous oscillations like those of Figure 23 are not. The problem extends to the simulation feedback approximation, and the one-step time delay required necessary to prevent a paradox in the system where the input draws on output loads information while simultaneously computing the same. The sensitivity was not apparent in the low inertia simulations since it is believed the system stability margins are quite large; the differential equation is not as "stiff" for low inertias.

Summary. The simulation runs quantify intuitive trends for vehicle response as a function of powertrain inertia. At one extreme (low drivetrain inertia) the vehicle reaches a speed of 45 MPH in (an unrealistic) time of about 27 seconds. At the other extreme it takes approximately 45 seconds to reach a speed of just over 25 MPH.

Shock load predictions, though not as severe as expected, were well within the bounds of the (extremely) conservative static loading case of Section 3.2. Oscillation of the steer drive output torque was also expected, but not to the extent the simulation suggested for a system with all inertias present.

3.5 Weight Estimate and Center of Gravity Location

For the prescribed engine compartment geometric constraints, it became clear that system weight and center of gravity changes were the Achilles heel of the adaptation of the GSD-10 concept to the AAV-7A1. To closely track and quantify specific problems, Lotus spreadsheets were created for weight and balance of candidate driveline arrangements. Figure 13 provides a overview of the four basic configurations considered in the present work.

Option 1 (see Figure 13) was the first concept considered. In this arrangement, a "short" commercial transmission was sought such that the engine would not move aft. The output of the transmission then was routed up and back over the transmission with the steer drive straddling the transmission. The weight and C.G. of the resulting design are summarized in Table 17. The high weight of this option is due, in part, to the multiple transfer cases and shafting needed to route power up and down within the geometric constraints.

Option 2 was conceived to reduce the weight of option 1. By switching the vertical positions of the transmission and steer drive units, the high-torque transfer cases were replaced with low torque transfer cases. The weight, C.G. and performance results are summarized in Table 17. Although the weight was reduced relative to option 1, the overall weight was not satisfactory.

Option 3 was explored briefly as a method of eliminating all vertical and lateral transfer cases. By keeping all components "in-line" the system weight can be minimized and the overall C.G. can be improved. This method was presented to DTNSRDC and was rejected as a potential modification for the AAV-7A1/A2. This option remains attractive for new amphibious vehicle concepts.

Option 4 minimizes the need for transfer cases while squeezing components within the existing confines of the engine compartment. The engine and transmission are arranged side by side with the steer drive unit located forward of the engine and transmission. The engine is rotated 180 degrees relative to its present orientation such that the engine output is facing aft. This engine/transmission geometry is used in the M-113 family of vehicles. Detailed drawings of this configuration are found in Appendix D. Detailed weight and center of gravity calculations were performed for this configuration.

3.5.1 AAV-7A1 Data Base. Detailed weight and center of gravity values were unavailable by "component" for the current configuration of the AAV-7A1. Top level breakdowns (e.g. "transmission") for the AAV-7A1 were obtained from PMS-310 as were component breakdowns for the earlier LVTPX-12 and LVTP-7 vehicles. To compensate for missing component data, the detailed breakdowns from earlier vehicles were used. A calibration of the present vehicle was obtained by adding an artificial weight and center of gravity entry in the spreadsheet, this brought the overall group total in line with those of the current vehicle. The resulting analysis is satisfactory for current purposes.

3.5.2 Weight and Balance Computations. Weight and center of gravity computations were performed for each of the configurations shown in Figure 13. Greater emphasis was placed on option 4 when it became clear that this option offered the minimum weight within geometric constraints specified by DTNSRDC. The following results are found in Appendix E:

Sheet 1 of 6 - LVTP-7 Weight and C.G., February 1971
Sheet 2 of 6 - AAV-7A1 Weight and C.G., Baseline
Sheet 3 of 6 - AAV-7A1/GSD-10, Power Train Without Transmission
Sheet 4 of 6 - AAV-7A1/GSD-10, Transmission
Sheet 5 of 6 - AAV-7A1/GSD-10, Weight and C.G.
Sheet 6 of 6 - AAV-7A1/GSD-10 Auxilliary Propulsion System

By comparing sheets 2 and 5, one notes that the GSD-10 installation has increased vehicle weight by 760 pounds and has resulted in a C.G. shift of 10 inches forward.

3.6 <u>Component Corrosion Control</u>

Use of the AAV-7A1 in salt water environments requires that new or modified components be subject to the same corrosion resistance standards the existing system enjoys. The proposed steer drive system is intrinsically corrosion resistant for the following reasons:

1. Only the external case of the steer unit is exposed to the atmosphere inside the engine compartment,
2. All internal components are immersed in oil or are splash lubricated.

The steer drive casing and chain drive transfer case are to fabricated with ASTM corrosion resistant aluminium alloys 5456 or 5083. Specifications for the automatic transmission call for a cast aluminium alloy "...meeting or exceeding SAE J453 #306...". Since the DDA 750 series uses a casting meeting the SAE J453 #306 standard there are no anticipated corrosion problems with the transmission. The oil pan is either cast iron or stamped steel (SAE 1010) and is typically treated during manufacture and assembly to combat corrosion on highway vehicles. Main hydraulic piping elements are made of stainless steel and all auxiliary components are corrosion resistant.

A unique challenge is presented by the brake system since air-cooled brakes traditionally employ cast, ductile iron discs. When exposed to a salt-water environment ductile iron corrodes quickly. When iron is heated - as the discs will be during typical braking conditions - the corrosion proceeds at an alarming rate. To eliminate the corrosion problem it is expected the discs will be cast phosphor bronze. Bronze is regularly used in salt water applications, in components such as ship propellors. This material will not compromise braking capabilities, but will result in a specialized item.

4.0 DATA ACQUISITION PLAN

Data acquisition plans were established for the GSD-10 steer drive system before and after its installation in the AAV-7A1. Analysis of the data would be used to check efficiency and performance predictions from the EDS and serve as a guideline for future refinements in the GSD-10 system adaptation. Of particular interest were the quantification of:

(a) improved steering response over the complete range of AAV-7A1 speeds,
(b) regenerative characteristics,
(c) skid-out capabilities,
(d) changes in gradability and tractive effort,
(e) waterjet performance.

Inclusion of a data acquisition plan as part of the EDS was for the purpose of (a) determining if any special test facilities would be required and (b) anticipating test requirements in advance so appropriate provision for the same could be incorporated (if possible) during the system design.

4.1 Data Acquisition During Fabrication and Installation

4.1.1 Data Collection During Fabrication. Anticipating some difficulty in backing out weight, size, and performance data once the steer unit has been installed, provision for these measurements will be made during the fabrication of the unit. Specifically, the following information is of interest and will be collected:

(a) exact steer unit weight with and without lubricating fluids present,
(b) center of gravity measurements (balance method),
(c) detailed geometric layout for "as-built" system drawings,
(d) mechanical efficiency tests under conditions simulating rectilinear and curvilinear motion,
(e) effective inertia measurements for conditions simulating rectilinear and curvilinear motion.

It is anticipated that the last two tests will require equipment not traditionally found in a fabrication house, and that such test-bench equipment may then require the services of an as-yet undesignated test sub-contractor.

4.1.2 Data Collection During Installation. Data collected during this phase will include:

(a) update to Engine/Transmission/Steer Drive installation instructions,
(b) manhours required to install each component,

(c) detailed parts list for installed system, and
(d) corrosion treatment activities.

4.2 Vehicle Tests

The second phase of the general test plan covers those integrated system tests related to AAV-7A1 system performance. The objective of these tests is to obtain the following information and/or verification:

(a) steering characteristics, available turn radii at different speeds, steering efficiency, and regeneration factor,
(b) full operation of all vehicle systems that were modified,
(c) proper operation of the vehicle in each mode of operation and safe transition between each mode of operation,
(d) power availability to the track and waterjet systems,
(e) fuel economy,
(f) braking and power regeneration capabilities, and
(g) tractive effort and available drawbar pull, both instantaneous and continuous.

4.2.1 Measurements to be Considered. Data for the performance analyses can be obtained from five basic tests.

DRAWBAR PULL MEASUREMENT

Purpose: To measure the tractive effort (TE) of the vehicle.

Use: Since the tractive effort to weight ratio is a measure of tracked vehicle performance, the measurement of drawbar pull allows the ratio to be computed and the transmission to be recognized as having the required torque capacity at low speed. Equipped with the HS-400, the vehicle presently has a tractive effort to weight ratio of 1.2.

Test Requirements: A serrated concrete surface is required such that the risers on the vehicle track can positively engage the ground. The tracks typically will begin to spin on smooth concrete (even with rubber track pads) at a tractive effort to weight ratio near 0.7. Reinforcement of the trailer hitch may be required on the vehicle. A load cell (typically hydraulic) is rigged in series with a towing chain. A second chain is used to anchor the first to a large object. A remote readout on the load cell is required to keep personnel well clear of the tensioned members.

Recommended Objective: Demonstrate a tractive effort to weight ratio of 1.2 (50,434 lb-ft @10.5 inch sprocket radius) so the GSD-10 adaptation is considered "as good as" modern tracked vehicle transmissions.

SPROCKET TORQUE MEASUREMENTS ON THE MOVE

Purpose: Display and record the torque being passed between the steer drive and the final drive as the vehicle maneuvers.

Use: To demonstrate that the steer drive is regenerative (i.e., that power is transferred from the inboard track to the outboard track when the vehicle enters a sharp turn at relatively high speeds). The efficiency of this regeneration should be a major strength of the steer drive.

Test Requirements: Requires several acres of open field and large areas of concrete where the vehicle can turn sharply at speeds of 20 to 45 MPH.

Required Instrumentation: Two shaft torque transducers with a capacity of 18,000 ft-lb to be installed between the steer drive and the final drive input flange. On-vehicle display of the analog torque signal is very useful for demonstrations. The permanent record may be kept on a strip chart recorder as the simplest means of data recording.

Conjugate Data Required: Shaft speed into each final drive, engine speed and engine torque are all required if a complete technical assessment is to be obtained. Furthermore, the sign of the torque on each shaft is needed to recognize (on the strip chart record) when the torque is positive and when it is negative. This would seem easy to construct if a yaw-accelerometer signal is included, but in practice it is not difficult to determine when the torque changes sign given the large oscillations in such signals.

ENGINE OUTPUT POWER

Purpose: To monitor and record the power delivered to the transmission in various situations. With the vehicle operating on flat surfaces, and knowing that the rolling resistance varies consistently (as a function of track tension and soil conditions) between 70 and 120 lbs/ton, it is possible to monitor the gross efficiency of the transmission. Knowing the engine power in or out of the transmission is also key to measuring steering performance.

Required Instrumentation: Torque transducer (1,000 ft-lb capacity) between the steer drive and the engine. Engine speed can be measured at the flywheel with a tooth counting sensor. The product of these signals can be used to display power.

SPROCKET SPEED

Purpose: Used in conjunction with sprocket torque measurement to generate a numerical value for power into or out of each final drive.

Instrumentation: Drill and tap the final drives to install a Hall-effect sensor.

VIDEO TAPING

Purpose: To document the vehicle's acceleration, turning performance, braking performance, etc. The video tapes may be the single most important data in terms of illustrating the benefits of the GSD-10 system.

4.2.2 Technical Support for Vehicle Tests. Several of the test to be considered are, in principle, easy to perform, but as a practical matter are difficult to instrument for. In particular, the addition of pre-calibrated torque transducer sections may not be geometrically permissable, and therefore require strain gauge rosettes to be mounted and calibrated on steer drive components prior to installation in the vehicle. This requires the testing facility to be selected well in advance so that testing considerations are made prior to steer drive construction and installation.

It is also recognized that different facilities will have different approaches to the measurements to be considered. As such, the ideal test facility would be one where Military Tracked Vehicles are already being tested. Preliminary discussions were held with Aberdeen Proving Ground (APG) personnel. It appears that APG is in an excellent position to provide the technical support for the desired vehicle tests. APG has the shop facilites to provide most of the necessary instrumentation (on and off-board the vehicle). Water and beaching tests are not possible at APG.

Reccomendations are that testing be scheduled through APG; close liaison with APG for planning and test approval during the design and fabrication phases is believed to lead to the most effective and efficient test option. There is significant value to a third party collecting data so its validity is not subject to interpretation prior to analysis.

5.0 FUTURE WORK

5.1 Overview

Advantages associated with the use of the Gleasman Steer Drive in the AAV-7A1 vehicle mission include improved steering at low speeds, improved shifting performance, high reliability, lower life cycle cost, and modular repair. These benefits are compromised by a power transmission weight increase of 23% relative to the existing HS-400 drivetrain. Since Marine Corps amphibious vehicles must perform well in water, design changes affecting weight and center-of-gravity are of significant concern.

Although the weight of the present version of the GSD-10 is a negative factor relative to existing transmissions, a major portion of the weight can be moved aft without impacting the use of the rear ramp of the vehicle. This is possible by moving the steer unit aft and splitting it between the port and starboard sides of the vehicle. Thus, the disadvantage associated with a relatively small weight increase may be offset by the improvement in the vehicle center-of-gravity.

Further work needs to be performed on the design analysis for the adaptation of the Gleasman Steer Drive to a rear-drive tracked amphibious vehicle. While adaptation would focus on the existing AAV-7A1 vehicle, the optimal application would be in future amphibious vehicles with no powertrain layout bias. Future vehicles will also require transmissions with higher power to weight ratios and increased performance capabilities. Adaptation of the Gleasman Steer Drive to these vehicle concepts is anticipated to be considerably easier and cost effective than with the upgrade of existing specialized military units.

5.2 Rear-Sprocket Drive Concept

A rear-sprocket driven amphibious vehicle represents a departure from the current front-sprocket approach. Such a departure requires a design/arrangement of the AAV be established for the purposes of a technology demonstration; this demonstration is not valid unless a configuration is ultimately acceptable from a mission standpoint. Although a preliminary study indicates the rear-drive system is the best system for optimizing a GSD-10 in the AAV family of vehicles, a more detailed design effort is needed to geometrically locate the steer assembly in the rear of the vehicle. It is proposed the engine and transmission remain in the front of the vehicle in an in-line configuration. A drive shaft tunnel will run from the front to the rear of the vehicle near the vehicle centerline. The design objective shall be to locate the steer drive in two separate parts in the rear of the vehicle. The existing final drives would be relocated to an aft position. This reconfiguration should accomplish the following:

a. the center of gravity of the vehicle would move aft by at least 6 inches.
b. the overall weight of the GSD-10 driveline would be minimized.
c. additional troop seating would be gained by moving the engine forward and rotating it 180 degrees. The transmission, which is only 28 inches in girth, would protrude through the engine firewall but would be segregated from the troop seating by its own cover. It is anticipated that this would open up seating space for an additional 5 personnel.

Implementation of a rear-drive concept permits a significant change in the approach to the waterjet drive system, making it more amicable to a high-speed vehicle concept.

5.3 Recommendation for Future Work

The present AAV system study provides an excellent base for the adaptation of the Gleasman Steer Drive to a rear-drive vehicle concept. It is recommended that future work focus on:

a. design integration of a rear-drive power train.
b. mission impact of AAV-7A1 rear-drive concept.
c. steer drive and transmission subsystem control.
d. AAV-7A1 performance study for new configuration.
e. weight estimates and center-of-gravity location.
f. identification of major components.

Extensive use of prior design results and the computer-aided design tools developed for the front-drive configuration could be applied to the rear-drive concept. Key points for investigation include the potentially adverse impacts of "lumps" in the floor of the troop compartment and potentially improved cargo and personnel capacity.

6.0 CONCLUSIONS

Several design problems were encountered in the adaptation of the Gleasman Steer Drive System to the AAV-7A1.

1. The gear reduction ratio of truck differentials tends to be large for AAV applications.
2. Overall system weight precludes a ""bolt-together"" GSD-10 concept.
3. A limited number of fully-automatic transmissions are commercially available as off-the-shelf items.
4. System weight is 740 lbs over HS-400.
5. Engine must be moved to allow power differential to be in-line with power differential; this causes "ripples" in cooling, fuel, and structural design.

Generally these problems extend from the distributed power train concept and the geometric restrictions within the existing engine compartment.

While the difficulties mentioned above focused on the tasks of torque and speed capabilities, a number of problems related to control were successfully solved.

1. A steer pump and motor were identified that are expected to provide excellent steering control.
2. A pump control valve has been located that yields a simple control system.
3. The use of a new torque converter allows integrated (off-the-shelf) control of the converter and transmission.
4. Waterjets were converted to hydraulic drive.

In summary, the following conclusions have been drawn from the Engineering Design Study:

1. The GSD-10 concept offers steering performance, gradability, and top speed, that are equal to or better than the most advanced tracked vehicle transmissions.

2. Approximately 70% of GSD-10 components can be purchased as commercially available items.

3. The GSD-10 weight and volume are inherently greater than the weight and volume of an integrated design in which all components are packaged together for minimum weight and volume.

4. Simple control functions for both steering and shifting are associated with the GSD-10; the simplicity and reliability of the control system is a major advantage relative to competing designs.

5. Weight and space are used more efficiently in the adaptation of the GSD-10 where the engine, transmission, and steer unit are in-line.

6. The GSD-10 offers air cooled brakes which cannot contaminate hydraulic oil or transmission lubricating oil. Although air cooled brakes increase system weight, this is counterbalanced by an improvement in transmission reliability.

7. The GSD-10 design allows front-engine, rear-steer- drive configurations which will shift the center of gravity of a Marine Corps Amphibious Assault Vehicle aft, improving the in-water trim of the vehicle.

8. The existing commercial production base for many of the GSD-10 components will lower the overall cost of the transmission; projections are the system will cost at least 30% less than existing military tracked vehicle transmissions.

9. The modular arrangement of the GSD-10 will allow field replacement of failed items without pulling the transmission. An exception may be the transmission itself; however, the transmission has a mean time between failures in excess of 5,000 hours of service.

10. The GSD-10 is more difficult to mount in the vehicle than is an integrated transmission. Additional attention to shock mounting, vehicle deflections, and alignment are required due to the multiple mounting points.

11. The surf mode requires a relatively complex hydraulic circuit to power both the waterjets and the tracks simultaneously. If the system is to be applied to a new vehicle design, consideration may be given to alternate hydraulic arrangements for this mode.

12. The addition of a two-speed final drive would cut the requirement for transmission torque dramatically. This would allow the use of a smaller, less-expensive, and lighter transmission such as the Renk Doromat series. Such two-speed final drives have been built and tested by the USMC.

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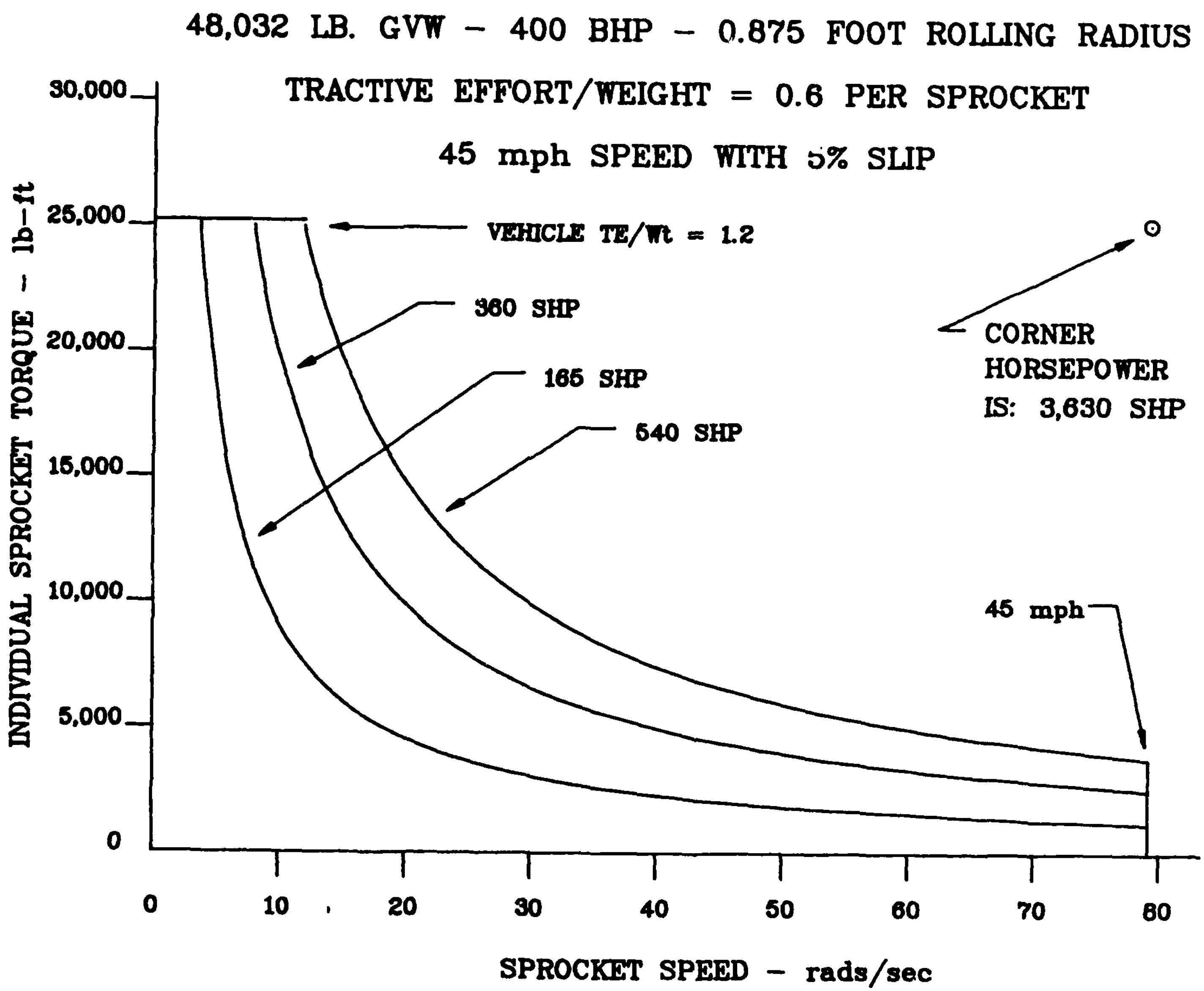


FIGURE 1.
SPROCKET SPEED/TORQUE LIMITS FOR AAV–7A1/GSD–10

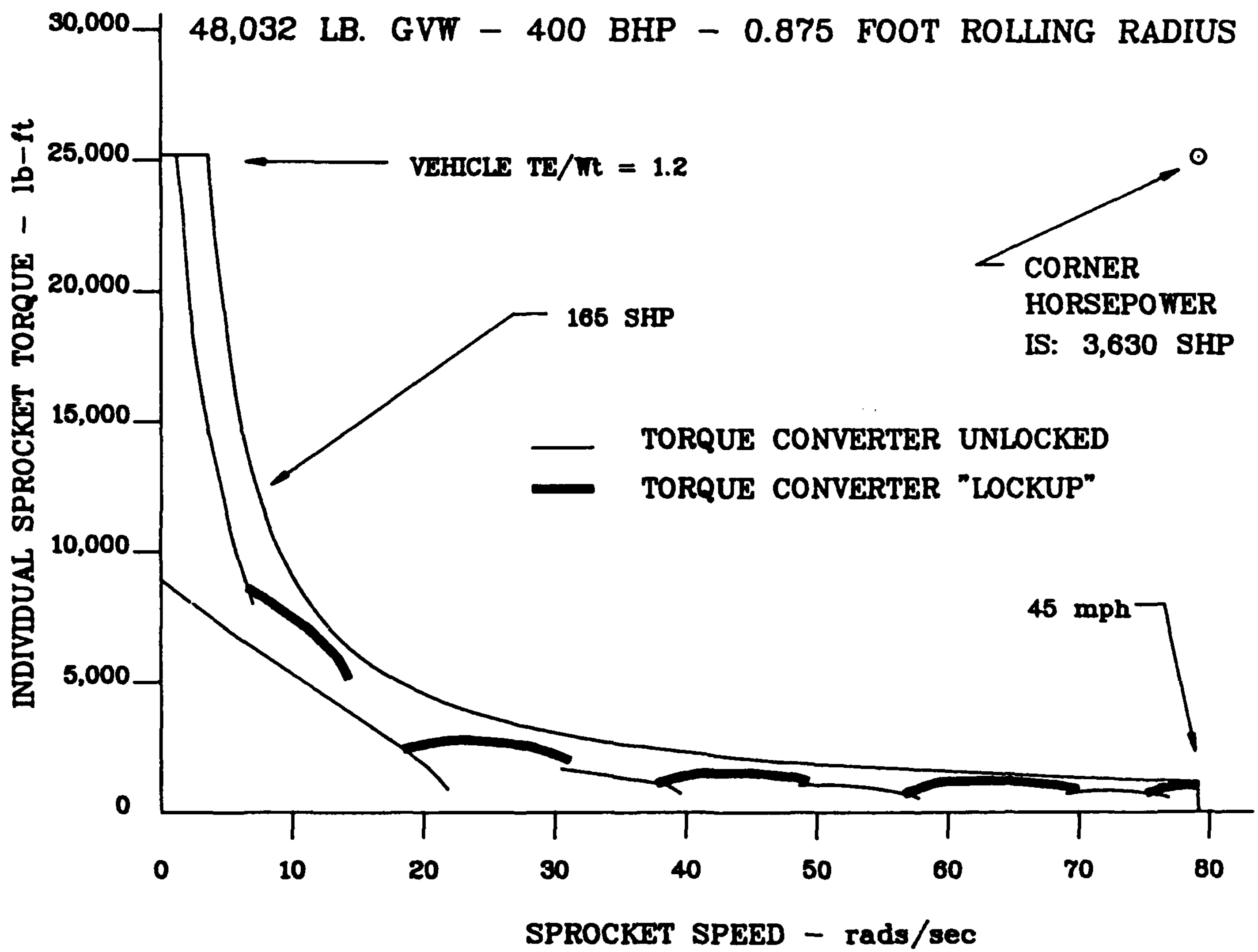


FIGURE 2.

TYPICAL 5-SPEED TRANSMISSION WITH TORQUE CONVERTER

STEERING MODE		INNER TRACK SPEED	INNER TRACK TORQUE	OUTER TRACK SPEED	OUTER TRACK TORQUE
RECTILINEAR MOTION		1	+	1	+
CURVILINEAR MOTION	PIVOT STEER	0	−	0<V<1	+
	COUNTER-ROTATION STEER	0<V<−1	−	0<V<1	+
	PLAIN STEER	0<V<1	+	1	+
	REGENERATIVE STEER	0<V<1	−	1	+

FIGURE 3. – REGIONS OF TRACKED VEHICLE MOTION AND STEERING

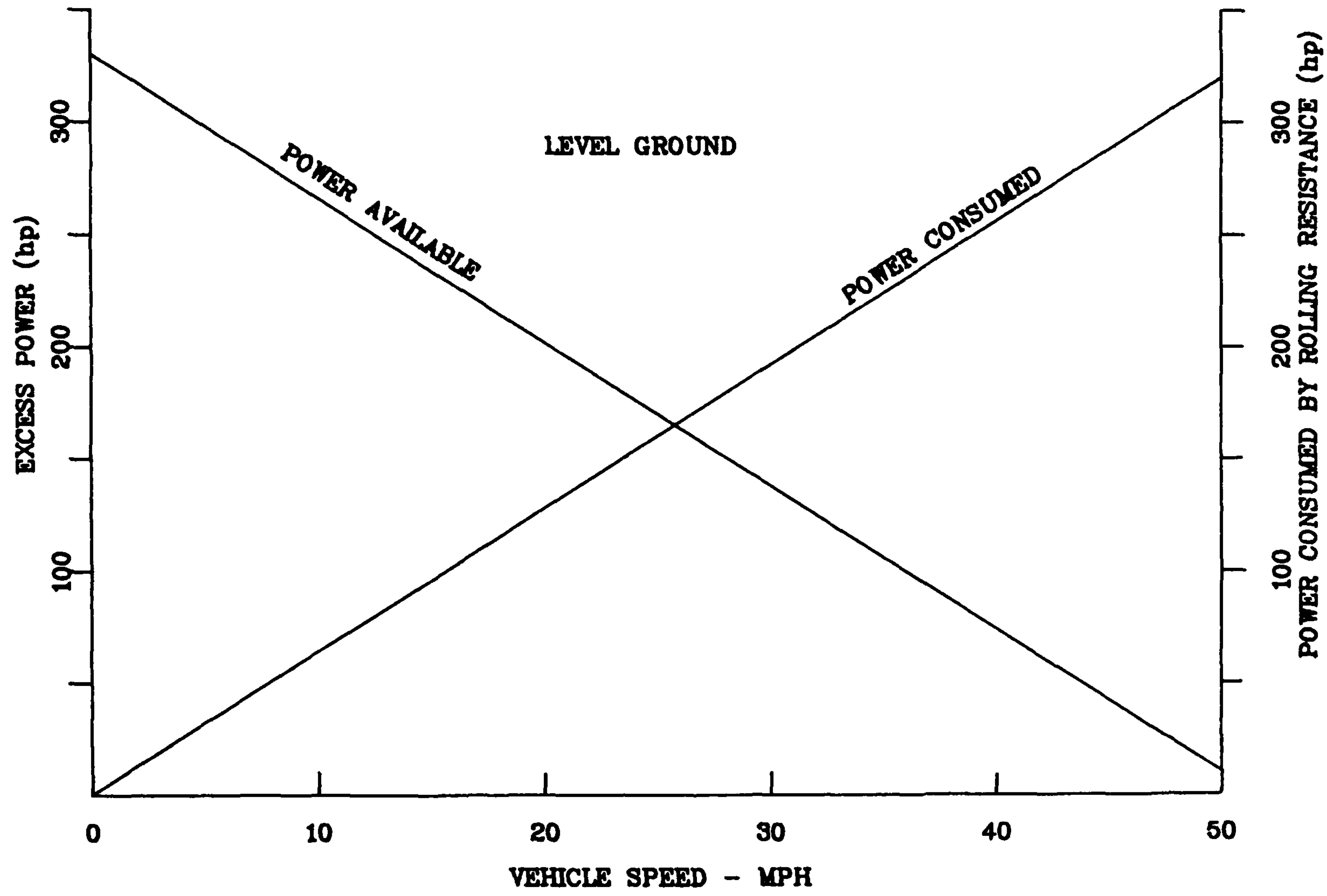
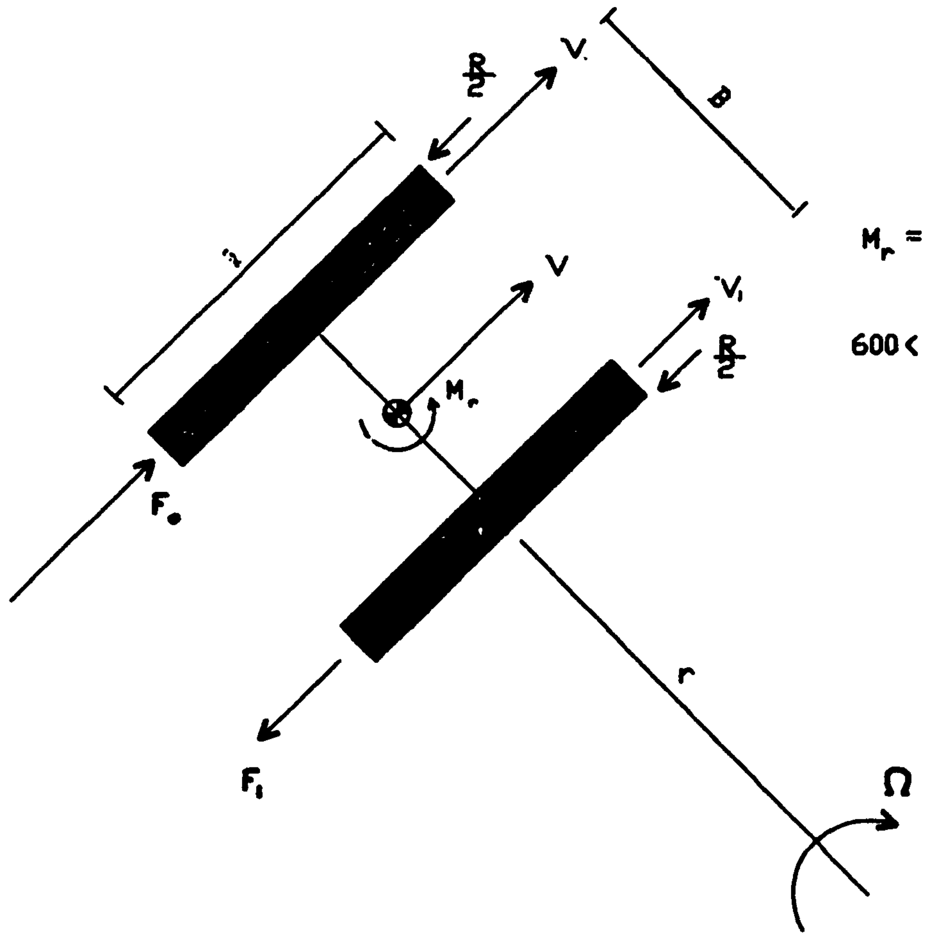


FIGURE 4. – EFFECT OF ROLLING RESISTANCE ON AAV-7A1 @ 100 LBS/TON



$$M_r = \frac{\mu_t W \ell}{4}$$

$$\mu_t = \frac{2 \int_0^{\ell/2} \mu_s \, d\ell}{\ell}$$

$$600 < \frac{R}{2} < 4{,}800$$

$$\ell = 13.3 \text{ Feet}$$

$$M_r = \frac{B}{2}(F_o + F_i)$$

FIGURE 5. – REGENERATIVE STEERING FORCES AND MOMENTS

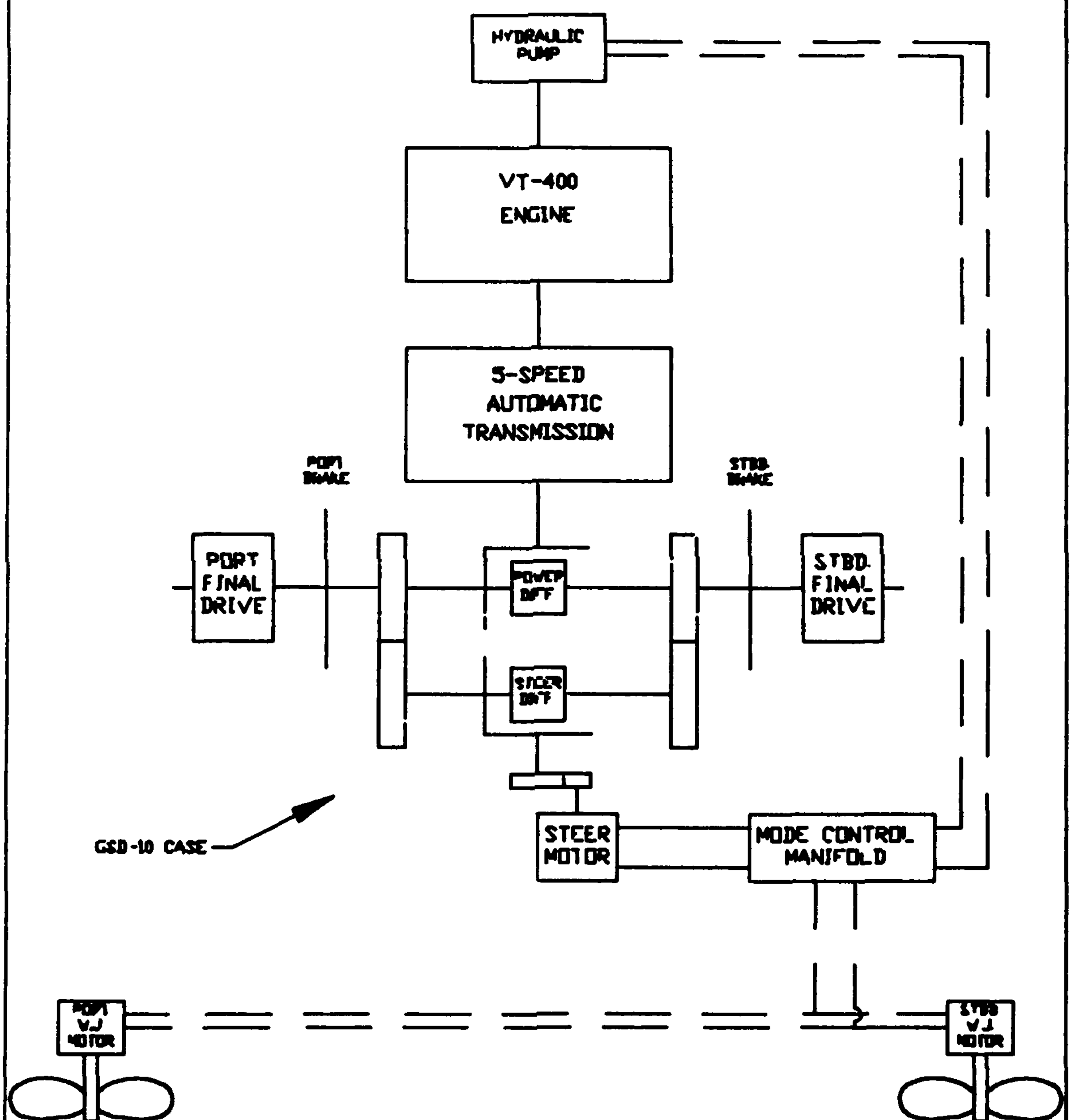


FIGURE 6.
AAV-7A1/GSD-10 SYSTEM SCHEMATIC

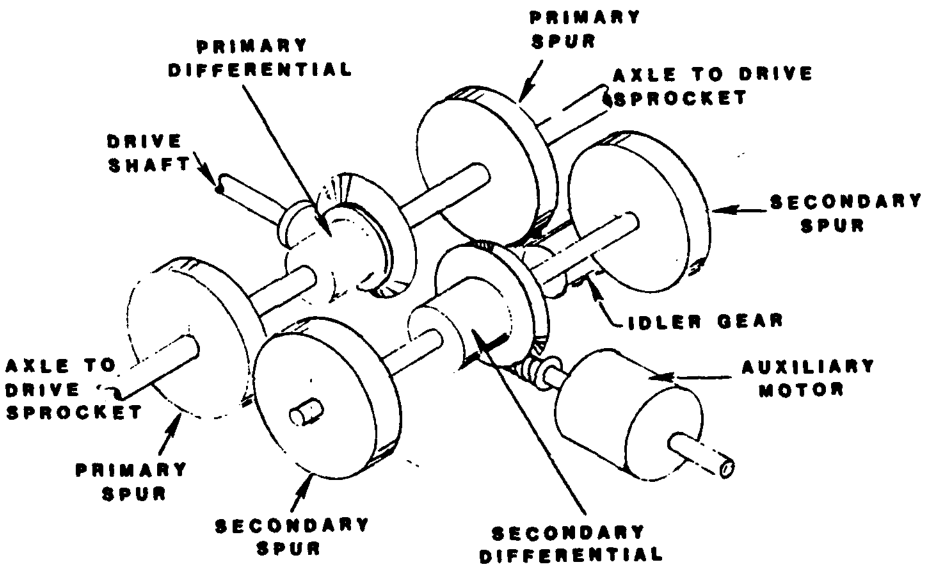


FIGURE 7 THE GLEASMAN STEER DRIVE SYSTEM (GSD-10)

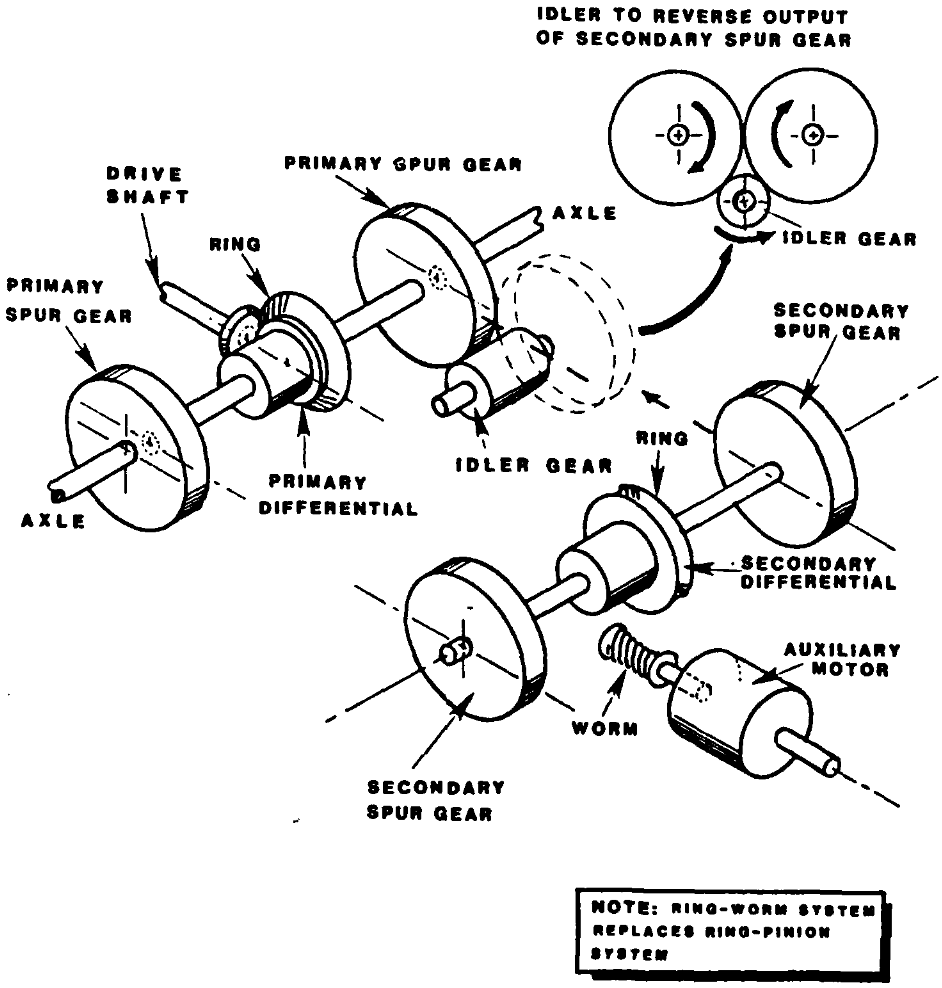


FIGURE 8 EXPANDED VIEW OF GSD-10 SYSTEM

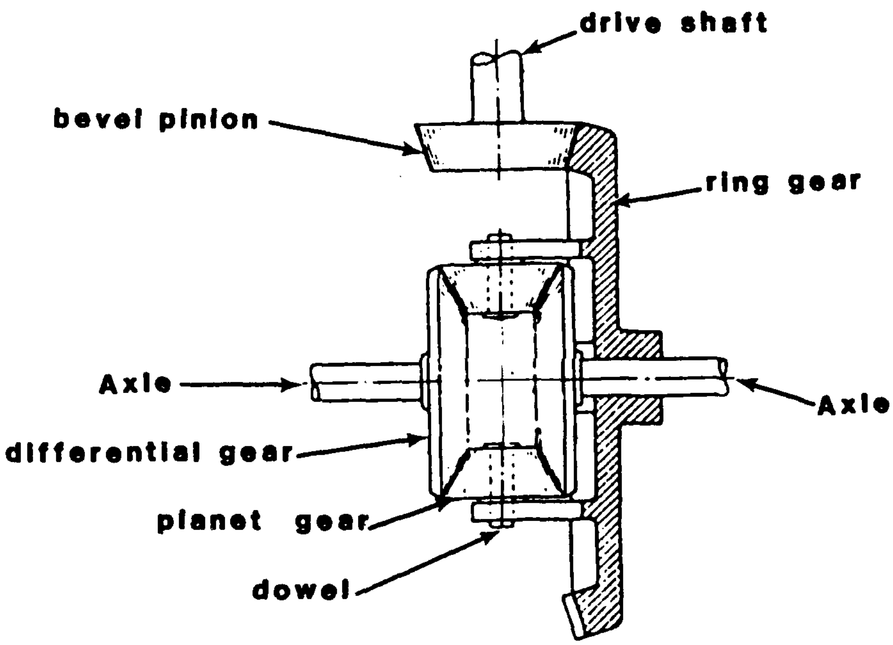
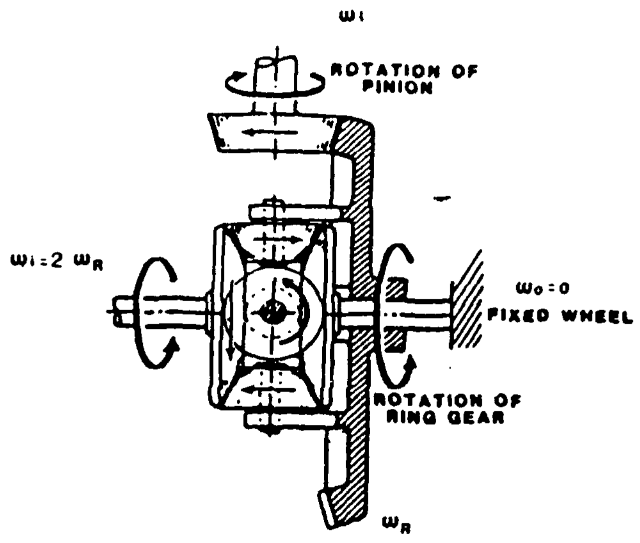
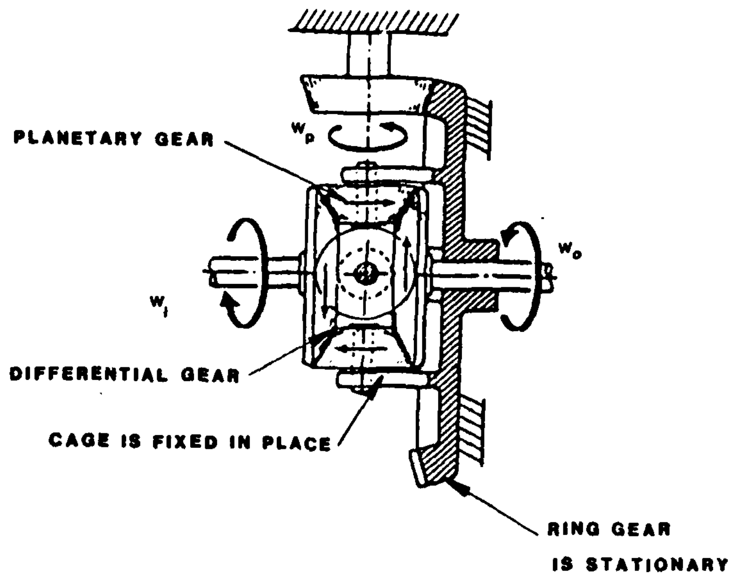


FIGURE 9 SIMPLIFIED VIEW OF DIFFERENTIAL



Kinematics with a fixed wheel



Kinematics of a fixed driveshaft.

FIGURE 10 GEAR DIFFERENTIAL CHARACTERISTICS

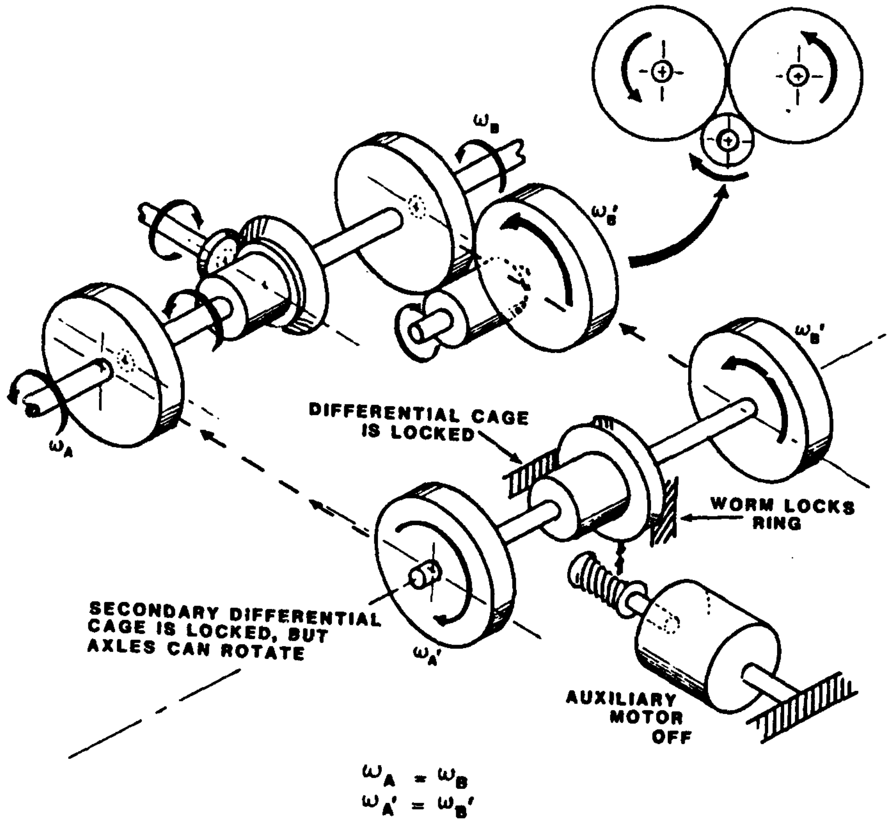


FIGURE 11 OPERATION IN RECTILINEAR MOTION

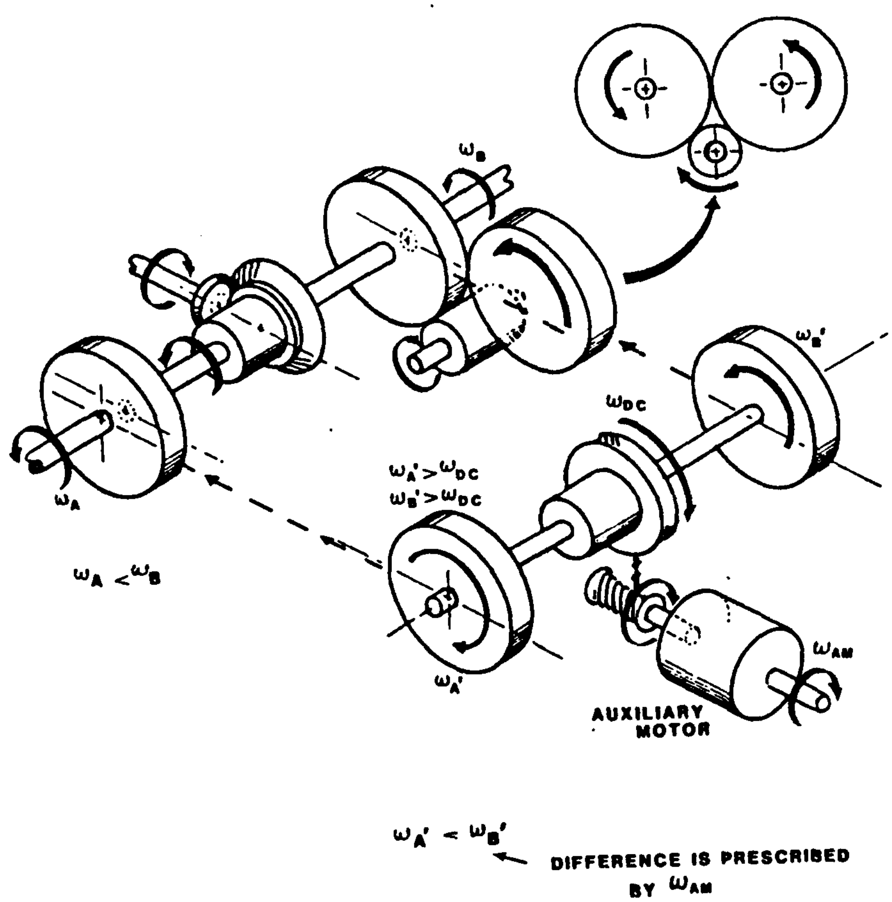


FIGURE 12 GSD-10 OPERATION IN <u>CURVILINEAR</u> MOTION

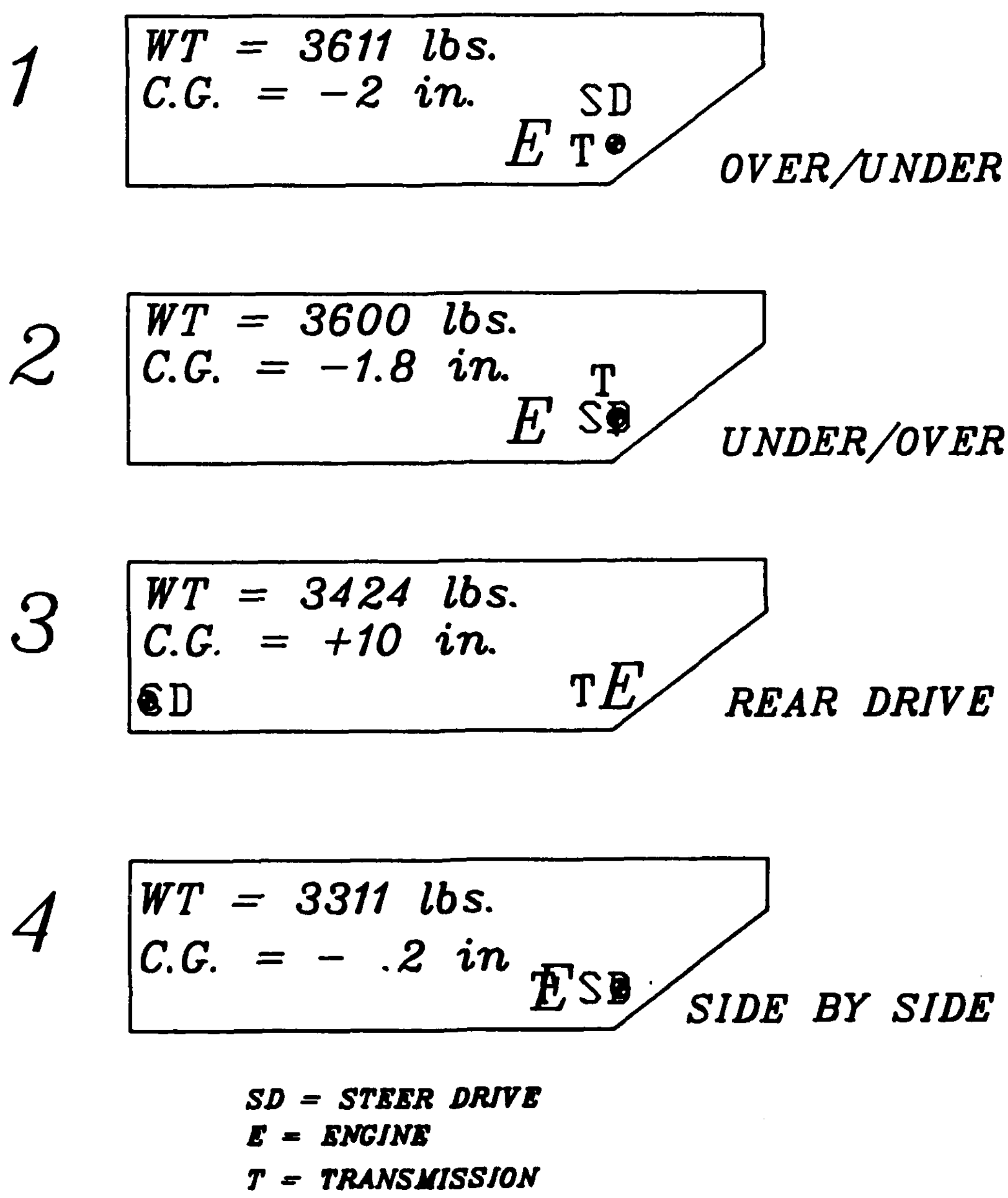


FIGURE 13.
AAV-7A1/GSD-10 WEIGHTS & CGs

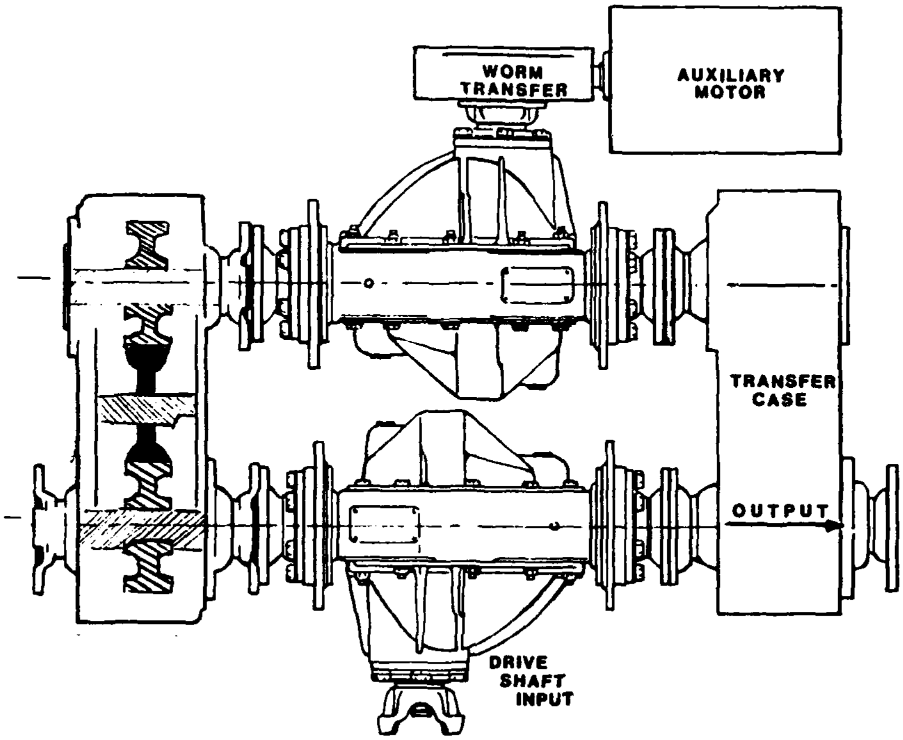
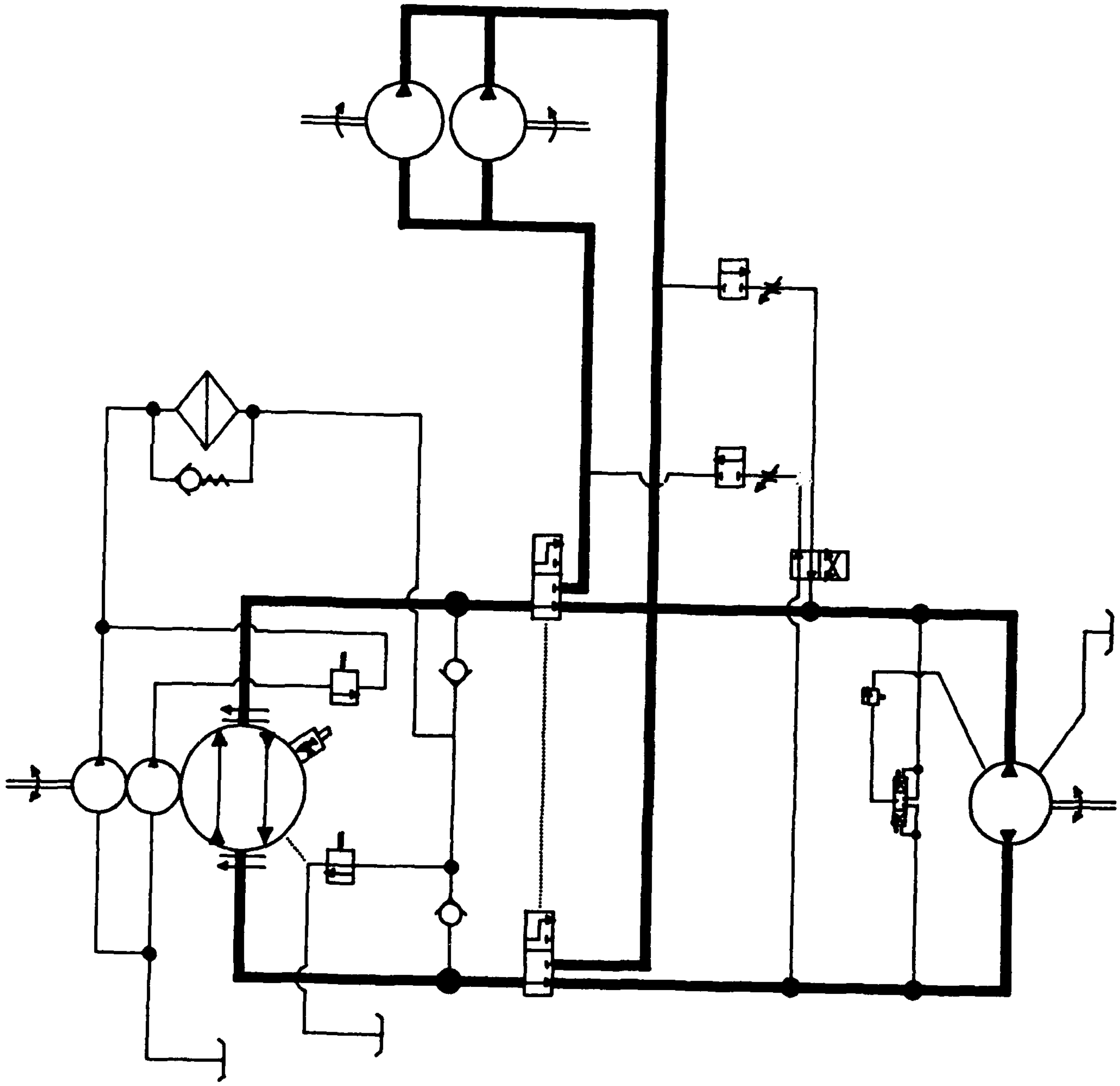


Figure 15 EARLY APPROACH TO GSD-10 PACKAGE



SIMPILFIED HYDRAULIC SCHEMATIC
FIGURE 16

Powertrain Subsystem Assembly Summary

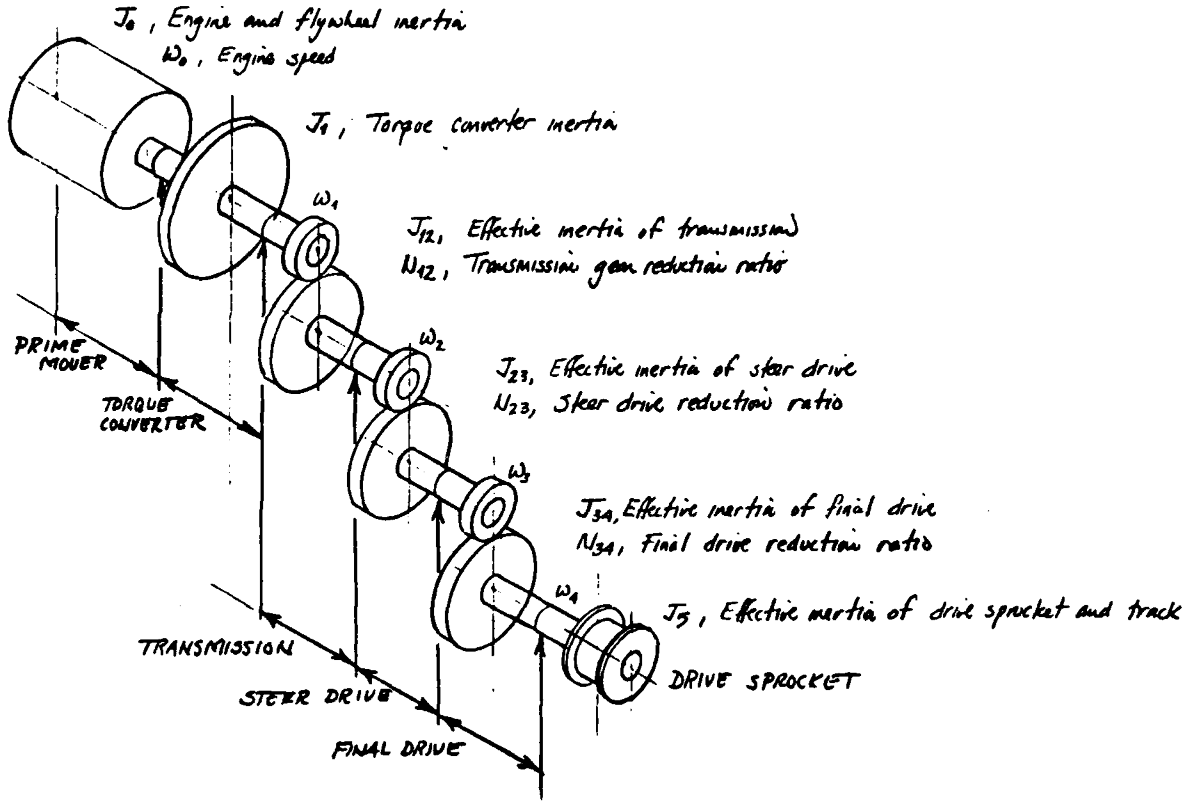


Figure 17. Generic representation of powertrain

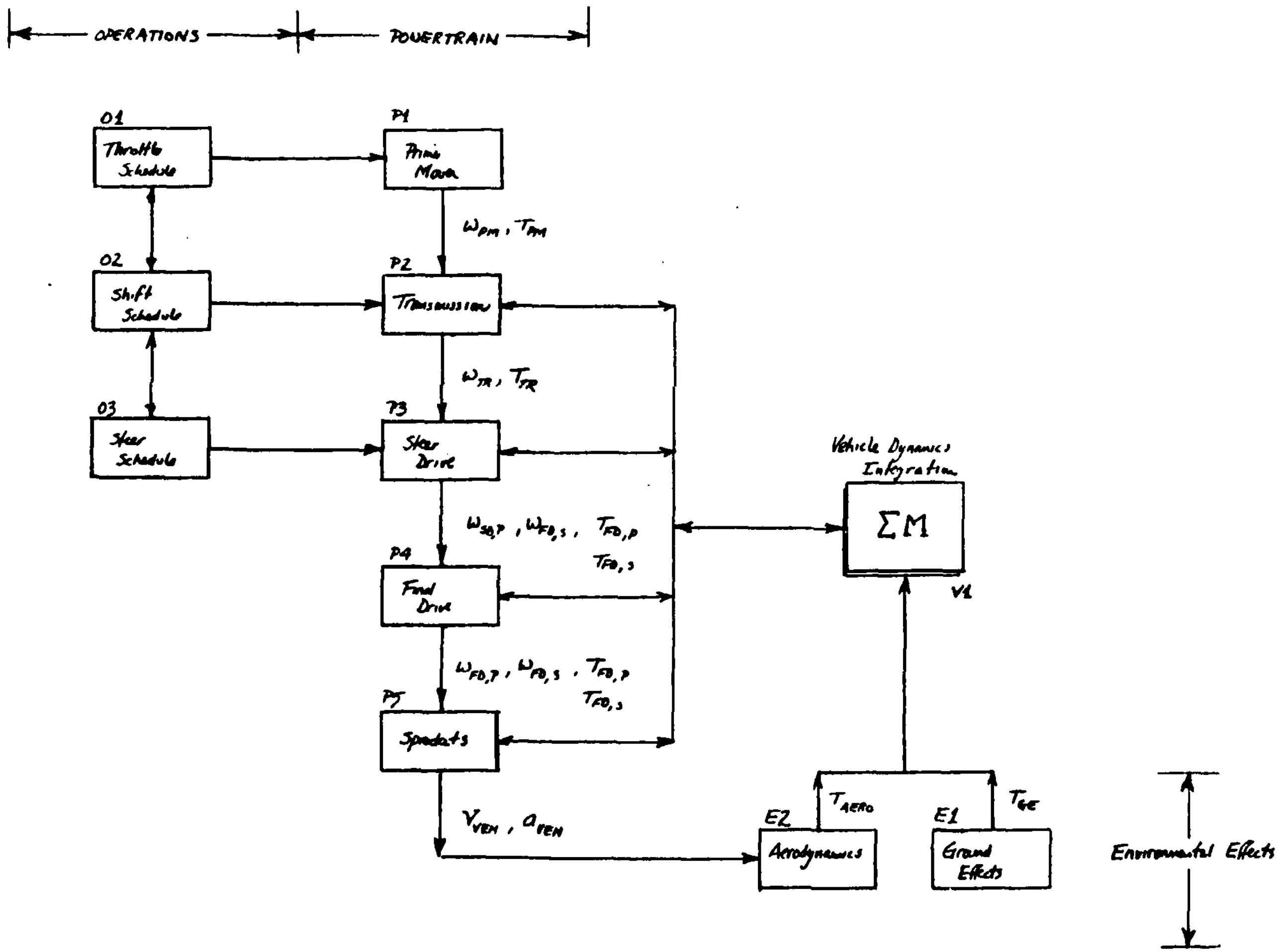
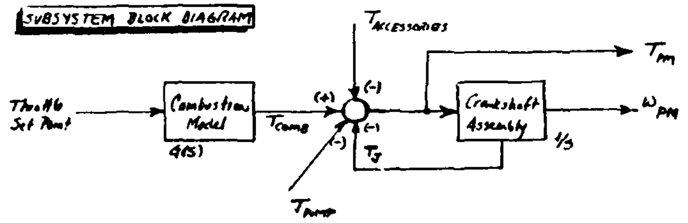
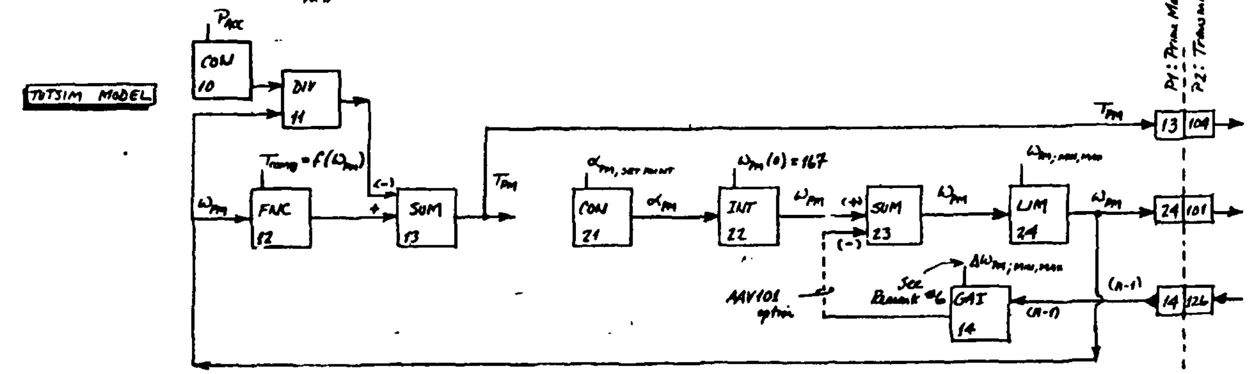


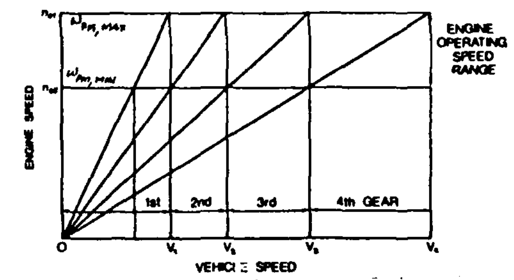
Figure 18. Block diagram for dynamic simulation

Figure 19a

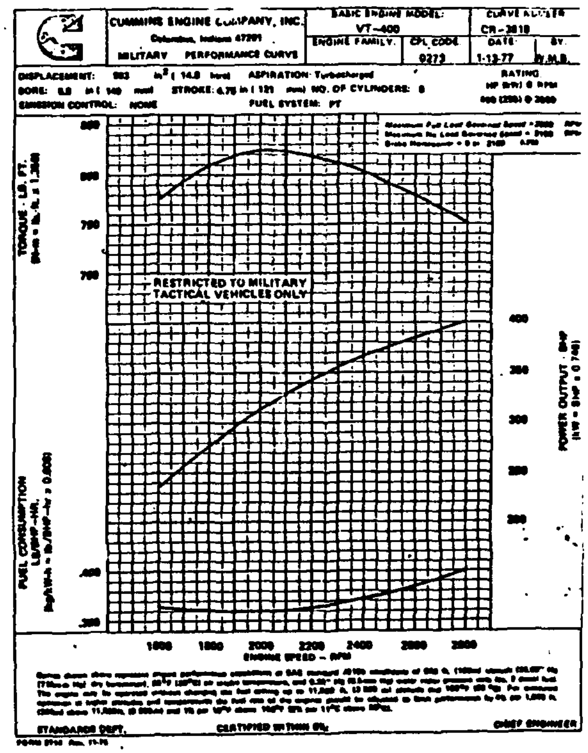
Module P1 : Prime Mover Simulation







Min and max engine speeds set in accordance with transmission shift schedule.



Remarks

1. Block 11 sets a prescribed engine acceleration set point; this overrides the need to account for engine inertia in simulation.
2. Blocks 14 and 23 dictate a drop in engine speed (to the lower engine operational limit) in accordance with the transmission shift schedule.
3. The engine is governed in block 24 so the maximum engine speed does not exceed $\omega_{E_{max}}$.
4. Engine output potential of 400 HP degraded by engine accessory loading; e.g., 47 Hp fan load and 15 Hp alternator load (47 + 15 = 62 Hp = 34,100 ft-lb/sec); Block 11.
5. Block utilization: Blocks 11-14 simulate combustion model and throttle control; 21-24 simulate crankshaft assembly; Torque converter pump load, T_{pump}, not required in present simulation.
6. Gain for block 14 is 0 for fixed shaft, and 26 rad/sec for shifting scenario.

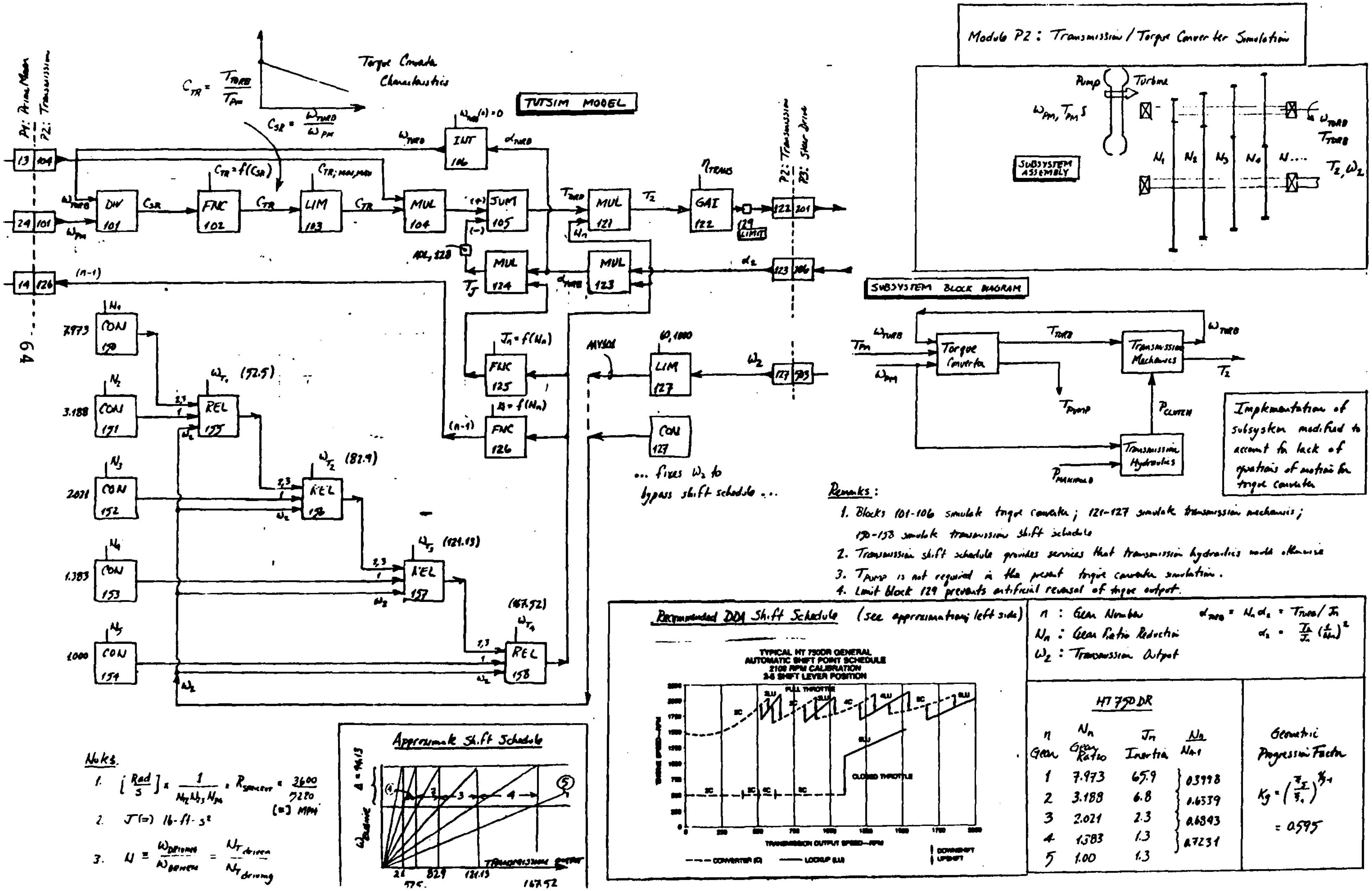
Notes

1. ω (=) rpm $\times \frac{2\pi\ rad}{rot} \times \frac{1\ min}{60\ sec} = 0.1047\ \frac{rad}{sec}$
2. $P = M\omega$ (=) Hp = 550 ft-lb/sec

Parameter	Value
α_{pm}	10 rad/s²
$\omega_{pm}(0)$	167.55 rad/s
$\omega_{pm\ min}$	167.55 rad/s (1600 rpm)
$\omega_{pm\ max}$	293.21 rad/s (2800 rpm)
T_E	ft-lb, see Table
μ	0.845
P_{acc}	34,100 ft-lb/sec
T_{comb}	see engine curves

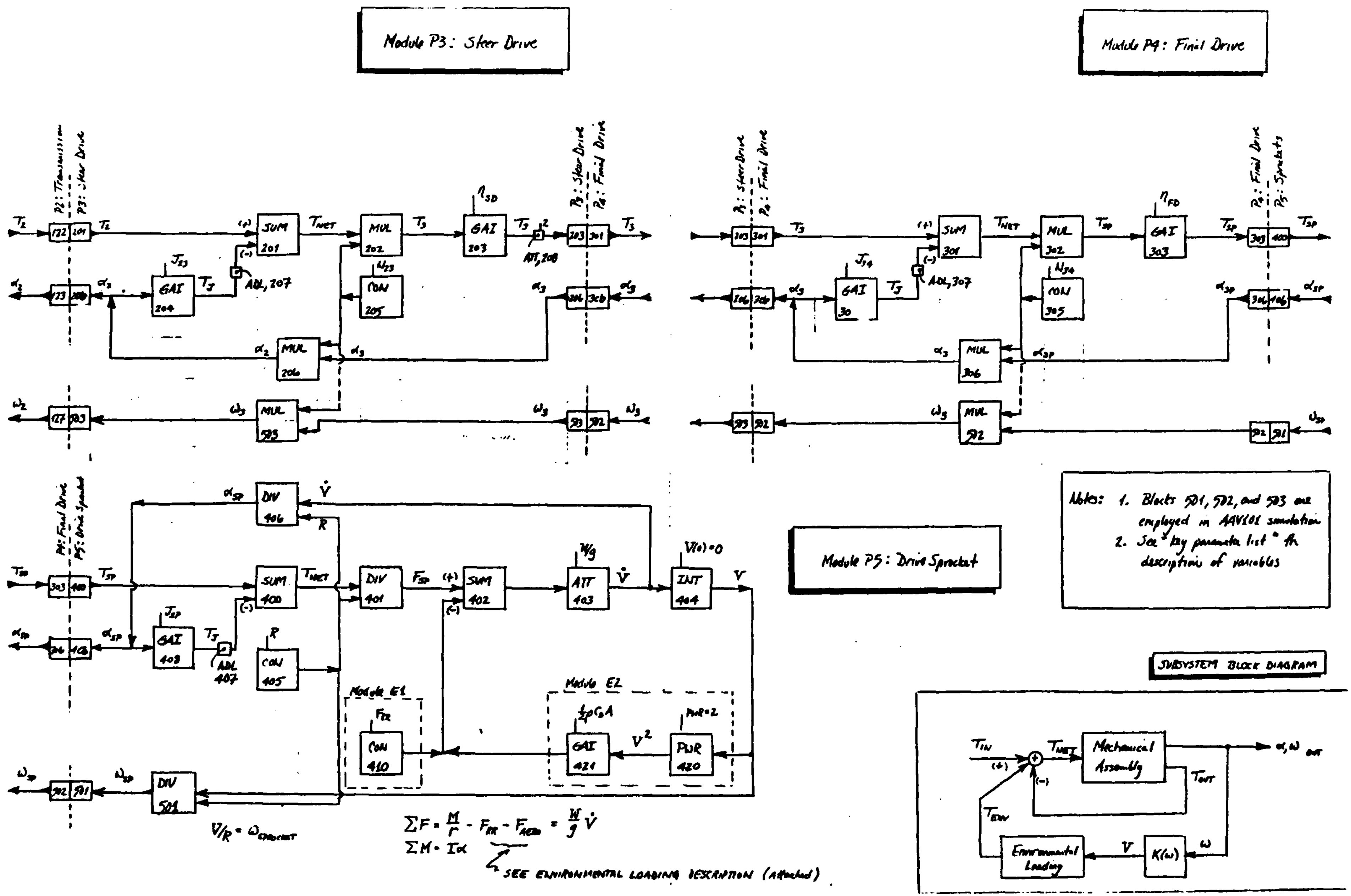
(See "Key parameter list" for more detail)

Figure 19b

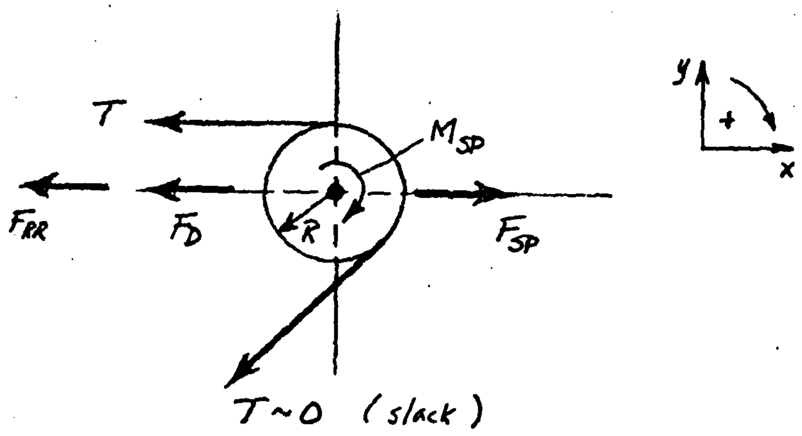


Module P2 : Transmission / Torque Converter Simulation
TUTSIM MODEL
SUBSYSTEM ASSEMBLY
SUBSYSTEM BLOCK DIAGRAM
Pump
Turbine
Torque Converter Characteristics
P1: Prime Mover
P2: Transmission
P2: Transmission
P3: Steer Drive
Torque Converter
Transmission Mechanics
Transmission Hydraulics
Implementation of subsystem modified to account for lack of equations of motion for torque converter.
... fixes ω₂ to bypass shift schedule ...
Remarks:
1. Blocks 101-106 simulate torque converter; 121-127 simulate transmission mechanics; 150-158 simulate transmission shift schedule
2. Transmission shift schedule provides services that transmission hydraulics would otherwise
3. T_Pump is not required in the present torque converter simulation.
4. Limit block 129 prevents artificial reversal of torque output.
Recommended DDA Shift Schedule (see approximation left side)
TYPICAL HT 750DR GENERAL AUTOMATIC SHIFT POINT SCHEDULE 2100 RPM CALIBRATION 2-5 SHIFT LEVER POSITION
FULL THROTTLE
CLOSED THROTTLE
TRANSMISSION OUTPUT SPEED—RPM
CONVERTER (C)
LOCKUP (LU)
DOWNSHIFT
UPSHIFT
Approximate Shift Schedule
Transmission Output
n : Gear Number
N_n : Gear Ratio Reduction
ω₂ : Transmission Output
HT 750 DR
n Gear | N_n Gear Ratio | J_n Inertia | N_n/N_n+1
1 | 7.973 | 65.9 | 0.3998
2 | 3.188 | 6.8 | 0.6339
3 | 2.021 | 2.3 | 0.6843
4 | 1.383 | 1.3 | 0.7231
5 | 1.00 | 1.3
Geometric Progression Factor
= 0.595
Notes.
1. [Rad/s] × 1/(N_T N_3 N_sp) = R_sensor = 3600/7280 (=) MPH
2. J (=) lb-ft-s²
3. N = ω_driving / ω_driven = N_T driven / N_T driving

Figure 19c. Modules P3-P5



Forces and Moments acting on vehicle / sprocket:



Vehicle balance:

$$\sum F_x = \frac{W}{g}\dot{V} = F_{SP} - F_D - F_{RR}$$

Sprocket balance:

$$\sum F_x = \frac{W_{SP}}{g}\dot{V} = F_{SP} - T$$

$$\sum M_0 = I_{SP}\alpha = M_{SP} - TR$$

<u>SIMPLIFICATION</u>

- Assume all vehicle weight assembled in W
- Ignore sprocket inertial loadings

Approximate balance:

$$\boxed{\sum F_x = \frac{W}{g}\dot{V} = \frac{M_{SP}}{R} - F_D - F_{RR}}$$

NOTE:
- Sprocket inertial loading can be re-entered through computation of NET M_{SP}; M_{SP} is that torque provided by drive train
- Rolling resistance shown co-linear with drag for convenience

FIGURE 19d
SPROCKET FORCE AND MOMENT APPROXIMATIONS

Environmental Loads

Module E1: Rolling Resistance

$F_{RR} = K W$

- K varies with speed, terrain conditions, and track maintenance but is assumed approx 90 lb/ton,

$F_{RR} \sim 2375$ lb

Module E2: Aerodynamic Resistance

$F_D = C_D q A$

- Assume flat-plate drag coefficient of 1.17 applies
- Approximate X-sectional area as $(129 \times 107)/144 = 95.85\ ft^2$
- Let air density be $\rho_{AIR} = 0.071\ lb_m/ft^3$
- Dynamic pressure

$$q = \tfrac{1}{2}\rho U^2$$
$$= \tfrac{1}{2}(0.071)U^2 \frac{1}{32.2} \; [=] \; \frac{lb_m}{ft^3}\,\frac{ft^2}{s^2}\left\{\frac{ft\cdot lb_m}{lb_f\cdot s^2}\right\}^{-1} \; [=] \; \frac{lb_f}{ft^2}$$
$$= 0.0011\,U^2\ lb_f/ft^2$$

Approximation for aero resistance is therefore

$$F_D = (1.17)(0.0011\ lb_f/ft^2)(95.85\ ft^2)\,U^2$$
$$= 0.1234\,U^2\ lb_f$$

FIGURE 19e
MODULES E1 AND E2, ENVIRONMENTAL LOADING

Conditions:
* No rolling resistance
* No final drive or steer drive inertia
* No track inertia

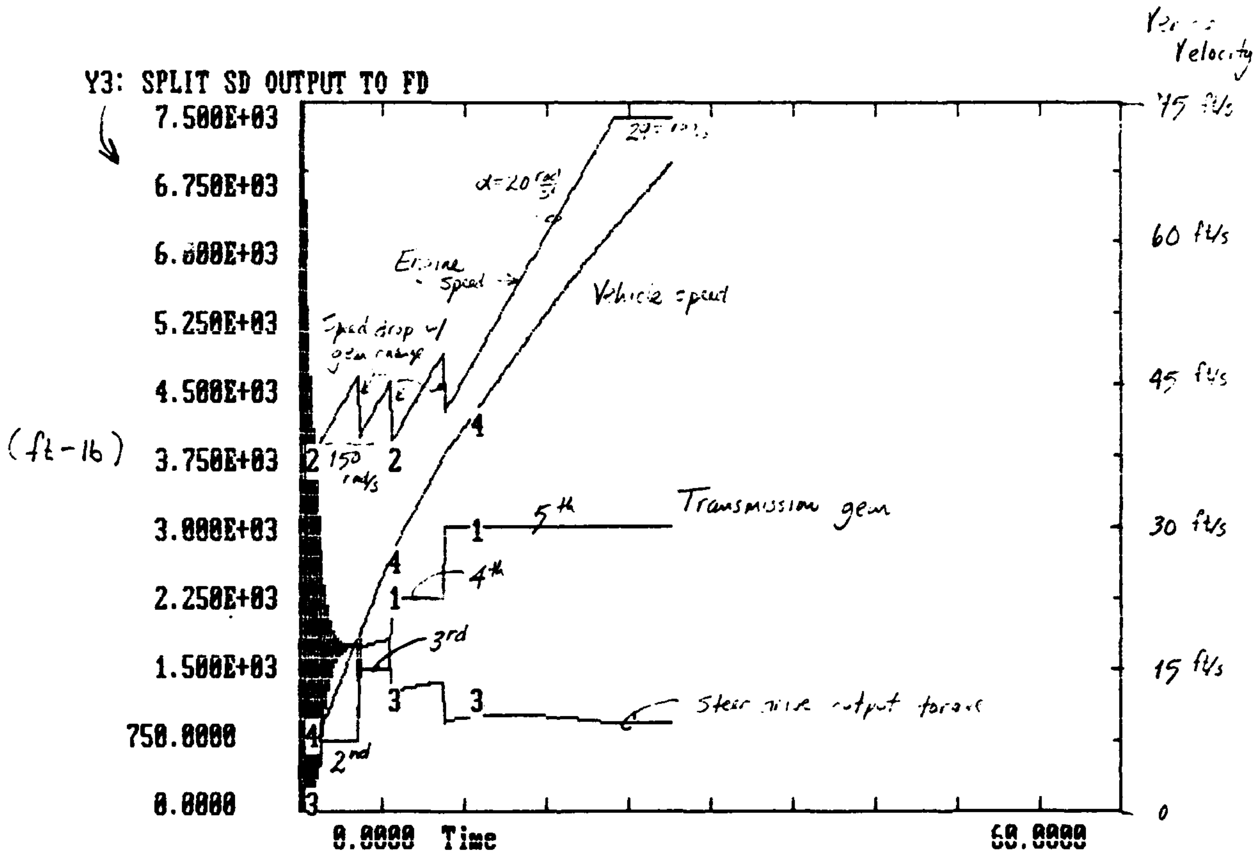


Figure 20 Dynamic response of system (Run #1)

Conditions:
* 80 lb/ton rolling resistance
* No final drive of steer drive inertia
* No track inertia

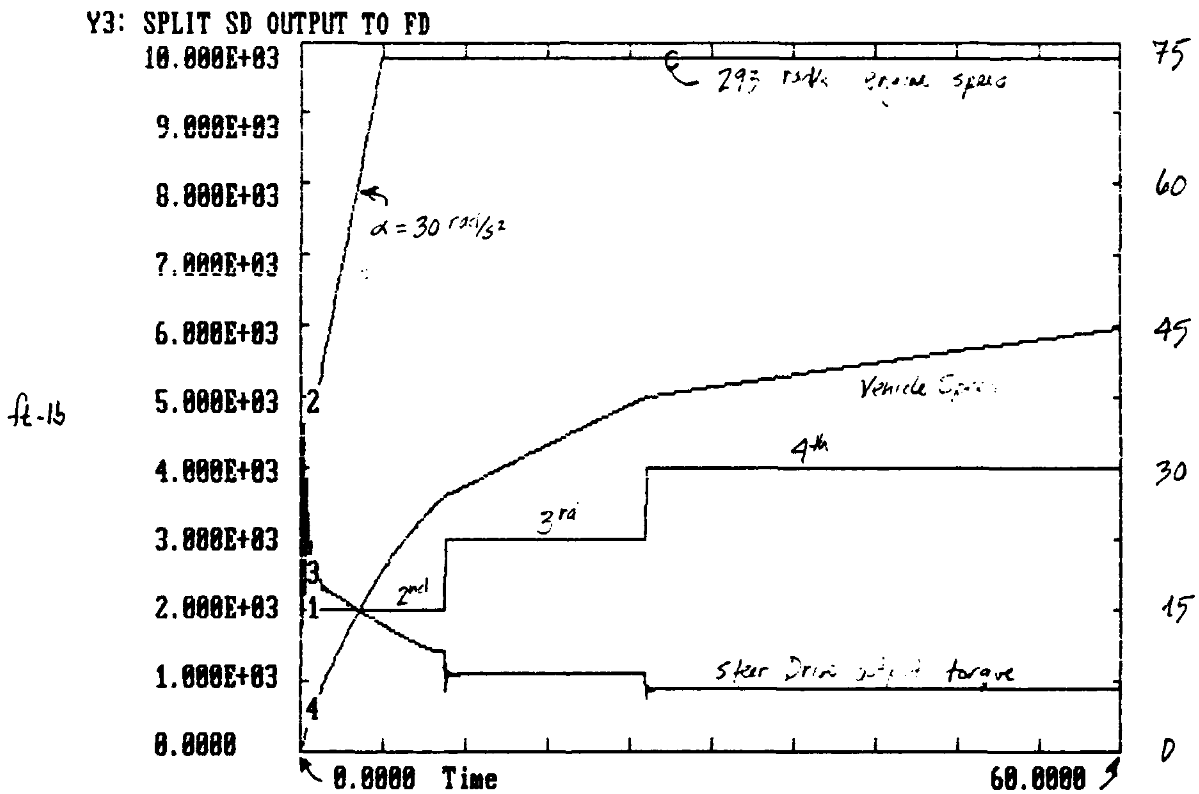


Figure 21. Dynamic response of system (Run #2)

Conditions:
* 80 lb/ton rolling resistance
* Final drive and steer drive inertia included
* No track inertia included

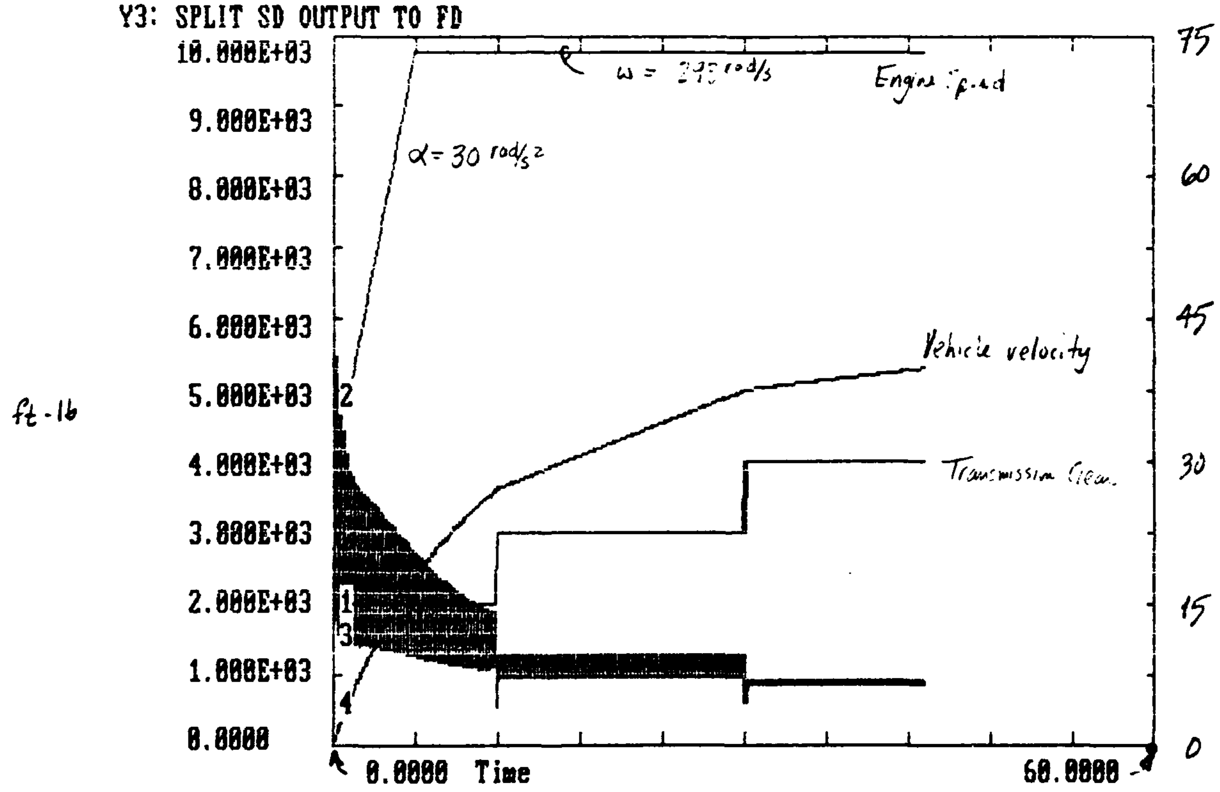


Figure 22. Dynamic response of system (Run #3)

Conditions:
* 80 lb/ton rolling resistance
* All system inertia approximations included.

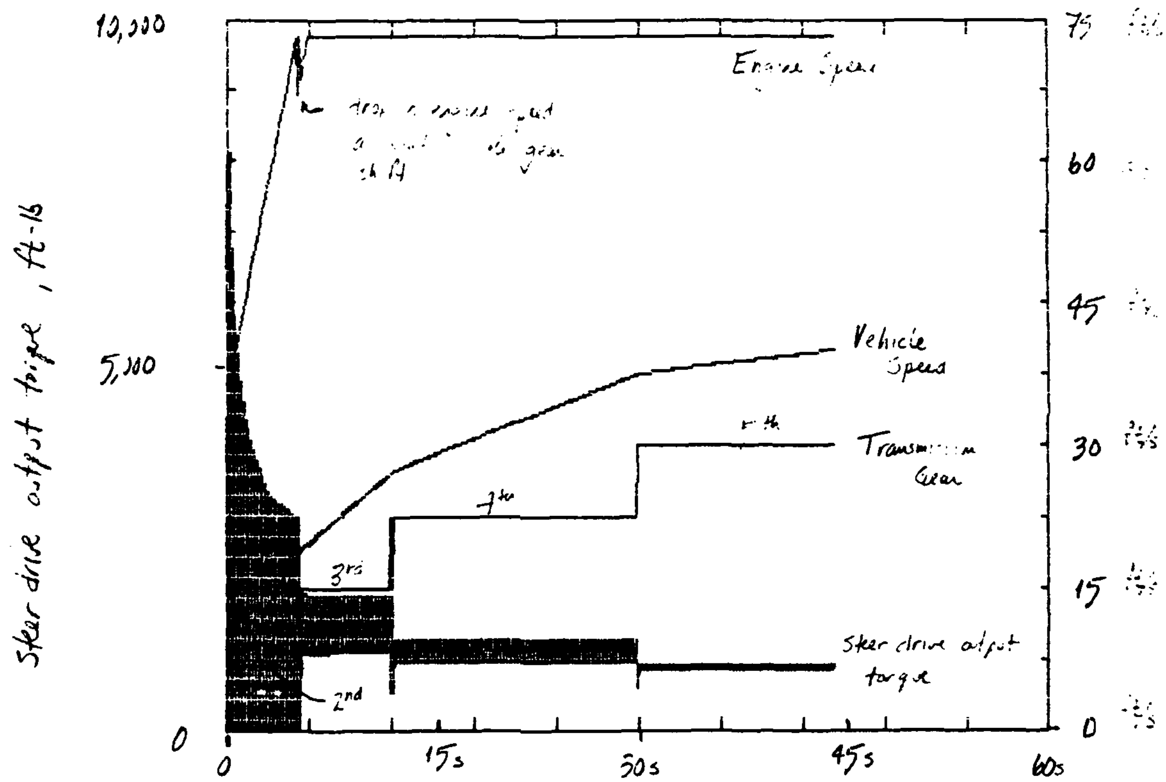


Figure 23. Dynamic response of system (Run #4)

TABLE 1
AAV-7A1 DESIGN & PERFORMANCE CRITERIA

General AAV-7A1 Data:

1. Maximum Vehicle Speed: 45 mph
2. Maximum Vehicle Gradability: 60 percent grade
3. Maximum Vehicle Tractive Effort to Weight Ratio: 1.2
4. Gross Vehicle Weight: 48,032 lbs (Combat Loaded)
5. Maximum fan horsepower (@2800 rpm): 45 hp
6. Accessory horsepower: 15 hp (hyd. & elec.)
7. Torque Converter: Twin Disk model 90192
   The torque ratio of the converter (i.e. output torque divided by input torque) varies from 3.3 at maximum slip to 0.98 at lockup.
8. HS-400-3A1 Gear Ratios & Efficiencies
   1st 8.27:1 eff. @ lockup is .93
   2nd 4.63:1 eff. @ lockup is .94
   3rd 2.25:1 eff. @ lockup is .95
   4th 1.27:1 eff. @ lockup is .96
9. Transmission Overall Maximum Split: 27.3
10. Final Drive Ratio: 3.06:1 (assumed efficiency is .95)
11. Sprocket Pitch Diameter: 21 inches
12. Rolling Resistance (measured values for P7)
    a. Extreme lowest value: 60 lbs/ton (concrete)
    b. Typical value: 100 lbs/ton (concrete)
    c. Worst value: 400 lbs/ton (soft sand)
13. Time to accelerate to speed (LVTP-7 with 8V-53T)
    a. 0-5 mph: 2.2 seconds
    b. 0-10 mph: 4.7 seconds
    c. 0-20 mph: 11.7 seconds
    d. 0-25 mph: 19.0 seconds
    e. 0-30 mph: 34.3 seconds
    f. 0-35 mph: 55 seconds
       (Note: times are somewhat better for VT-400 with HS-400 transmission improvements)
14. The rotational inertia of the driveline (including engine) is treated as an "added mass" term in the vehicle linear kinetic energy. At 40 mph the added mass is taken to be 10% of the vehicle mass. At 5 mph the added mass is taken to be 20% of the vehicle mass. The relationship between 5 and 40 mph is assumed to be linear. As speed approaches zero, the "added mass" term is assumed to remain at 20%.

TABLE 2

TECHNICAL PERFORMANCE OBJECTIVES FOR THE GSD-10

1. Maximum Vehicle Speed: 45 mph
2. Maximum Vehicle Gradability: 60 percent grade
3. Maximum Vehicle Tractive Effort to Weight Ratio: 1.2
4. Maximum Track Tractive Effort to Weight Ratio:
   a. Rectilinear: 0.6 Maximum T.E./Wt. Ratio per track
   b. Curvilinear: 0.5 T.E./Wt. Ratio per track from 20 mph to 45 mph.
5. Minimum Skid-Out Speed: 20 mph
6. Maximum Skid-Out Speed: 45 mph
7. Maximum CounterRotation Rate: 45 Degrees per Second
8. Braking Requirement: 4 Successive Crash Stops from 45 mph.
9. Time to Accellerate to 20 mph: Less than 11 Seconds
10. Overall Transmission Split: 22:1 or larger
11. Desired Driveline Weight: Less Than 2,500 lbs.
12. Desired Water Speed: 8.5 mph
13. Required Transmission Modes: Land, Surf, Water
14. Maximum Differential Sprocket Torque: 25,000 ft-lbs.
15. 95th Percentile Maximum Differential Sprocket Torque 15,000 ft-lbs. at speeds above 10 mph.

TABLE 3

```
CALCULATION OF INNER AND OUTER TRACK SPEED VERSUS SKID-OUT TURN RADIUS
     ASSUME A FLAT SURFACE
     ASSUME COEFFICIENT OF FRICTION IN LATTERAL DIRECTION IS 0.7
     THE COMBAT LOADED WEIGHT OF THE AAV-7A1 IS 48,031.6 lbs.
     THE COMBAT LOADED MASS OF THE AA1491.6645 SLUGS
     THE CENTRIPETAL FORCE IS: MV^2/R
     SKID-OUT OCCURS WHEN: MV^2/R = 0.7W
     OR: R=MV^2/.7W
       R IS THE TURN RADIUS IN FEET
       M IS THE VEHICLE MASS
       V IS THE VEHICLE VELOCITY IN FEET PER SECOND
       W IS THE VEHICLE WEIGHT
     ASSUME THAT INNER AND OUTER TRACKS EXPERIENCE A 5% SLIP IN TURNS
     THE INBOARD TRACK SLIP IS NEGATIVE (BRAKING)
     THE OUTBOARD TRACK SLIP IS POSITIVE (POWERING)
     THE VEHICLE VELOCITY IS: 2*PI*R/T
       T IS THE PERIOD OF ROTATION
     THE SEPARATION BETWEEN INNER AND OUTER TRACKS IS: 106 INCHES
     THE CIRCLE SCRIBED BY THE INNER TRACK HAS A RADIUS: R-4.41667 ft
     THE CIRCLE SCRIBED BY THE OUTER TRACK HAS A RADIUS: R+4.41667 ft
       Ri IS THE VELOCITY OF THE INNER TRACK
       Ro IS THE VELOCITY OF THE OUTER TRACK
     THE VELOCITY OF THE INNER TRACK IS: Vi=1.05*(2*PI*Ri/T)
     THE VELOCITY OF THE OUTER TRACK IS: Vo=1.05*(2*PI*Ro/T)
       Vi IS INNER TRACK VELOCITY INCLUDING SLIP IN fps
       Vo IS THE OUTER TRACK VELOCITY INCLUDING SLIP IN fps
     THE SPROCKET PITCH RADIUS IS 10.5 INCHES
     THE ROLLING CIRCUMFERENCE OF THE DRIVE SPROCKET IS: 5.50 FEET
     THE ANGULAR VELOCITY OF THE INNER SPROCKET IS: VAi=2*PI*(Vi/5.5)
     THE ANGULAR VELOCITY OF THE OUTER SPROCKET IS: VAo=2*PI*(Vo/5.5)
       VAi IS THE ANGULAR VELOCITY OF THE INNER SPROCKET IN rad/sec
       VAo IS THE ANGULAR VELOCITY OF THE OUTER SPROCKET IN rad/sec
     THE DIFFERENTIAL SPROCKET SPEED IS: Vd=VAo-VAi
     T=6.2832*R/V
     Vi=1.05*(6.2832*Ri/T)
       =1.05*(R-4.41667)*V/R
     Vo=1.05*(R+4.41667)*V/R
     VAi=6.2832*Vi/5.5
        =1.1424*(1.05*(R-4.41667)*V/R)
        =1.19952*(R-4.41667)*V/R
     VAo=1.19952*(R+4.41667)*V/R
```

TABLE 4

DEFINITION OF CONDITIONS AT SKID-OUT FOR COEF. OF FRICTION OF 0.7

VEHICLE SPEED mph	VEHICLE SPEED fps	SKID-OUT TURN RADIUS feet	INNER SPROCKET SPEED rad/sec	OUTER SPROCKET SPEED rad/sec	DIFFER. SPROCKET SPEED rad/sec	DIFFER. SPROCKET SPEED rpm	ANGULAR SPEED RATIO Ks
1	1.4666666	0.09543527	-79.6595	83.178140	162.83768	1554.981	-1.0441
2	2.9333333	0.38174110	-37.1908	44.228014	81.418844	777.4908	-1.1892
3	4.4	0.85891748	-21.8617	32.417502	54.279229	518.3272	-1.4828
4	5.8666666	1.52696441	-13.3175	27.391895	40.709422	388.7454	-2.0568
5	7.3333333	2.38588189	-7.48728	25.080248	32.567537	310.9963	-3.3497
6	8.8	3.43566992	-3.01403	24.125583	27.139614	259.1636	-8.0044
7	10.266666	4.67632851	0.683808	23.946335	23.262526	222.1402	35.0190
8	11.733333	6.10785765	3.897012	24.251723	20.354711	194.3727	6.22315
9	13.2	7.73025733	6.787125	24.880202	18.093076	172.7757	3.66579
10	14.666666	9.54352757	9.451075	25.734844	16.283768	155.4981	2.72295
11	16.133333	11.5476683	11.95054	26.753969	14.803426	141.3619	2.23872
12	17.6	13.7426797	14.32664	27.896455	13.569807	129.5818	1.94717
13	19.066666	16.1285616	16.60785	29.133836	12.525976	119.6139	1.75421
14	20.533333	18.7053140	18.81451	30.445775	11.631263	111.0701	1.61820
15	22	21.4729370	20.96151	31.817362	10.855845	103.6654	1.51789
16	23.466666	24.4314306	23.06005	33.237413	10.177355	97.18635	1.44134
17	24.933333	27.5807947	25.11868	34.697375	9.5786875	91.46951	1.38133
18	26.4	30.9210293	27.14405	36.190597	9.0465382	86.38787	1.33327
19	27.866666	34.4521345	29.14142	37.711826	8.5704046	81.84114	1.29409
20	29.333333	38.1741103	31.11497	39.256862	8.1418844	77.74908	1.26167
21	30.8	42.0869566	33.06812	40.822303	7.7541756	74.04674	1.23449
22	32.266666	46.1906734	35.00365	42.405368	7.4017131	70.68098	1.21145
23	33.733333	50.4852608	36.92385	44.003757	7.0798995	67.60790	1.19174
24	35.2	54.9707188	38.83065	45.615555	6.7849036	64.79090	1.17473
25	36.666666	59.6470473	40.72564	47.239153	6.5135075	62.19926	1.15993
26	38.133333	64.5142464	42.61020	48.873190	6.2629880	59.80699	1.14698
27	39.6	69.5723160	44.48547	50.516504	6.0310255	57.59191	1.13557
28	41.066666	74.8212562	46.35247	52.168103	5.8156317	55.53506	1. 2546
29	42.533333	80.2610669	48.21203	53.827130	5.6150927	53.62006	1.11646
30	44	85.8917482	50.06491	55.492841	5.4279229	51.83272	1.10841

TABLE 5
CANDIDATE DIFFERENTIALS FOR THE GSD-10

		DIFFERENTIALS	
COMPANY	DANA	ROCKWELL	LEYLAND
MODEL	70-X	9100	
WEIGHT	270	386	
REDUCTION, MIN	5.43	5.42	1.18
MAX OUTPUT TORQUE PER SHAFT	5,000 FT-LB	8,200 FT-LB	8,000

TABLE 6

SPEED AND TORQUE REQUIREMENTS FOR DIFFERENTIAL INTERFACE WITH FINAL DRIVES AND TRANSMISSION

```
**********************************
SEACO, Inc.:  GSD-10 Steer Drive System
POWER TRAIN COMPONENT INTERFACE ROUTINE
**********************************

VEHICLE: AAV-7A1
   Gross Vehicle Weight:            48031.0 lbs
   Final Drive Sprocket Radius:       10.5 inches

Transmission:    DDA HT-750
Differential:    Ideal Version

VT903 ENGINE
Maximum torque output:          825 ft-lbs
Maximum output speed:          2800 rpm

REQUIREMENTS FOR 45 MPH TOP SPEED

Maximum sprocket speed            720.000    RPM
Final drive ratio                   3.060        MFG SPEC
Final drive input speed          2203.200    RPM
Differential output speed        2203.200    RPM
Differential gear reduction         1.271        MFG SPEC
Differential input speed         2800.047    RPM
Trans-Diff gear step-up             1.000        COMPUTED
Transmission output speed        2800.000    RPM
Trans. ratio, high gear             1.000        MFG SPEC
Engine output speed, max rpm     2800.000        MFG SPEC

******************************************
*   Trans/Diff. Gear Step-up:        1.000   *
*   System Torque Capability:        1.390   *
******************************************

REQUIREMENTS FOR MAX TORQUE

Sprocket Torque:
   (a) TE/W Ratio                    1.200
   (b) TE                        57637.200 lbs
   (c) Sprocket Torque           50432.550 ft-lbs Total
                                 25216.275 ft-lbs per side
Final drive input:
   Efficiency          0.99
                                  8323.851 ft-lbs
Differential OUTPUT torque:
   Efficiency          0.99
                                  6615.729 ft-lb ea. shaft
Differential INPUT torque:       13231.459 ft-lb
Transmission OUTPUT torque:
   Efficiency          0.92
                                 14382.261 ft-lbs
Transmission STR:      3.040               MFG SPEC
Low gear ratio:        7.973               MFG SPEC
Torque Amplifier:     24.238               MFG SPEC
Required engine output:           593.379 ft-lbs
Req'd overall torqe ratio:         17.433
System Torque C-pability:           1.390
```

TABLE 6 (a)
LEYLAND-A DIFFERENTIAL

```
************************************
SEACO, Inc.:  GSD-10 Steer Drive System
POWER TRAIN COMPONENT INTERFACE ROUTINE
************************************
```

VEHICLE: AAV-7A1

Gross Vehicle Weight:	48031.0 lbs
Final Drive Sprocket Radius:	10.5 inches

Transmission:	DDA HT-750
Differential:	Leyland-A

VT903 ENGINE

Maximum torque output:	825 ft-lbs
Maximum output speed:	2800 rpm

REQUIREMENTS FOR 45 MPH TOP SPEED

Maximum sprocket speed	720.000	RPM	
Final drive ratio	3.060		MFG SPEC
Final drive input speed	2203.200	RPM	
Differential output speed	2203.200	RPM	
Differential gear reduction	1.180		MFG SPEC
Differential input speed	2599.776	RPM	
Trans-Diff gear step-up	0.928		COMPUTED
Transmission output speed	2800.000	RPM	
Trans. ratio, high gear	1.000		MFG SPEC
Engine output speed, max rpm	2800.000		MFG SPEC

```
**********************************************
*  Trans/Diff. Gear Step-up:         0.928    *
*  System Torque Capability:         1.390    *
**********************************************
```

REQUIREMENTS FOR MAX TORQUE

Sprocket Torque:			
(a) TE/W Ratio		1.200	
(b) TE		57637.200	lbs
(c) Sprocket Torque		50432.550	ft-lbs Total
		25216.275	ft-lbs per side
Final drive input:			
Efficiency	0.99		
		8323.851	ft-lbs
Differential OUTPUT torque:			
Efficiency	0.99		
		7125.365	ft-lb ea. shaft
Differential INPUT torque:		14250.730	ft-lb
Transmission OUTPUT torque:			
Efficiency	0.92		
		14382.261	ft-lbs
Transmission STR:	3.040		MFG SPEC
Low gear ratio:	7.973		MFG SPEC
Torque Amplifier:	24.238		MFG SPEC
Required engine output:		593.379	ft-lbs
Req'd overall torqe ratio:		17.433	
System Torque Capability:		1.390	

TABLE 6 (b)
LEYLAND-B DIFFERENTIAL

```
######################################
SEACO, Inc.:  GSD-10 Steer Drive System
POWER TRAIN COMPONENT INTERFACE ROUTINE
######################################

VEHICLE: AAV-7A1
   Gross Vehicle Weight:            48031.0 lbs
   Final Drive Sprocket Radius:        10.5 inches

Transmission:     DDA HT-750
Differential:     Leyland-B


REQUIREMENTS FOR 45 MPH TOP SPEED

Maximum sprocket speed            720.000     RPM
Final drive ratio                   3.060         MFG SPEC
Final drive input speed          2203.200     RPM
Differential output speed        2203.200     RPM
Differential gear reduction         1.230         MFG SPEC
Differential input speed         2709.936     RPM
Trans-Diff gear step-up             0.968         COMPUTED
Transmission output speed        2800.000     RPM
Trans. ratio, high gear             1.000         MFG SPEC
Engine output speed, max rpm     2800.000         MFG SPEC


###############################################
#   Trans/Diff. Gear Step-up:          0.968    #
#   System Torque Capability:          1.390    #
###############################################


VT903 ENGINE
Maximum torque output:                825 ft-lbs
Maximum output speed:                2800 rpm


REQUIREMENTS FOR MAX TORQUE

Sprocket Torque:
   (a) TE/W Ratio                     1.200
   (b) TE                         57637.200 lbs
   (c) Sprocket Torque            50432.550 ft-lbs Total
                                  25216.275 ft-lbs per side
Final drive input:
   Efficiency            0.99
                                   8323.851 ft-lbs
Differential OUTPUT torque:
   Efficiency            0.99
                                   6835.716 ft-lb ea. shaft
Differential INPUT torque:        13671.432 ft-lb
Transmission OUTPUT torque:
   Efficiency            0.92
                                  14382.261 ft-lbs
Transmission STR:        3.040              MFG SPEC
Low gear ratio:          7.973              MFG SPEC
Torque Amplifier:       24.238              MFG SPEC
Required engine output:             593.379 ft-lbs
Req'd overall torqe ratio:           17.433
System Torque Capability:             1.390
```

TABLE 6 (c)
DANA-SPICER DIFFERENTIAL

```
************************************
SEACO, Inc.:  GSD-10 Steer Drive System
POWER TRAIN COMPONENT INTERFACE ROUTINE
************************************

VEHICLE: AAV-7A1
   Gross Vehicle Weight:            48031.0 lbs
   Final Drive Sprocket Radius:        10.5 inches

Transmission:    DDA HT-750
Differential:    DANA-SPICER
```

```
VT903 ENGINE
Maximum torque output:          825 ft-lbs
Maximum output speed:          2800 rpm
```

REQUIREMENTS FOR 45 MPH TOP SPEED

Maximum sprocket speed	720.000	RPM	
Final drive ratio	3.060		MFG SPEC
Final drive input speed	2203.200	RPM	
Differential output speed	2203.200	RPM	
Differential gear reduction	5.430		MFG SPEC
Differential input speed	1.20E+04	RPM	
Trans-Diff gear step-up	4.273		COMPUTED
Transmission output speed	2800.000	RPM	
Trans. ratio, high gear	1.000		MFG SPEC
Engine output speed, max rpm	2800.000		MFG SPEC

```
**********************************************
*  Trans/Diff. Gear Step-up:          4.273   *
*  System Torque Capability:          1.390   *
**********************************************
```

REQUIREMENTS FOR MAX TORQUE

```
Sprocket Torque:
   (a) TE/W Ratio                     1.200
   (b) TE                         57637.200 lbs
   (c) Sprocket Torque            50432.550 ft-lbs Total
                                  25216.275 ft-lbs per side
Final drive input:
   Efficiency           0.99
                                   8323.851 ft-lbs
Differential OUTPUT torque:
   Efficiency           0.99
                                   1548.422 ft-lb ea. shaft
Differential INPUT torque:         3096.844 ft-lb
Transmission OUTPUT torque:
   Efficiency           0.92
                                  14382.261 ft-lbs
Transmission STR:       3.040               MFG SPEC
Low gear ratio:         7.973               MFG SPEC
Torque Amplifier:      24.238               MFG SPEC
Required engine output:             593.379 ft-lbs
Req'd overall torqe ratio:           17.433
System Torque Capability:             1.390
```

TABLE 7
CANDIDATE TRANSMISSIONS FOR THE GSD-10

	TRANSMISSIONS	
COMPANY	DDA	RENK
MODEL	HT750DR	874A
WEIGHT	940	642
SIZE	41 X 22.5 X 25.6	29.7 X 21 X 20
RATED, HP	425	315
TORQUE, FT-LB	1300	885
GEAR SPLIT	~8:1	~6:1

TABLE 8 - STEPS IN SELECTING HYDRAULIC PUMP & MOTOR

APPROACH TO SIZING STEER MOTOR AND PUMP:

1. COMPUTE VEHICLE TURN RADIUS AT "SKID-OUT-SPEED" FOR A COEFICIENT OF FRICTION OF 0.7
2. COMPUTE DIFFERENTIAL SPROCKET SPEED VS. SKID-OUT LIMITS
3. SET RANGE OF PERFORMANCE OVER WHICH SKID-OUT IS ATTAINABLE
4. SET LIMIT FOR COUNTER-ROTATION EQUAL TO MAXIMUM DIFFERENTIAL SPEED FOUND IN THE SKID-OUT PERFORMANCE
5. DETERMINE MAXIMUM SPEED OF CANDIDATE STEER MOTORS
6. DIVIDE MAXIMUM MOTOR SPEED BY MAXIMUM DIFFERENTIAL SPROCKET SPEED TO OBTAIN THE TOTAL REDUCTION RATIO FROM MOTOR TO SPROCKET
7. MULTIPLY MAXIMUM MOTOR TORQUE BY RATIO FROM 6. TO GET MAXIMUM DIFFERENTIAL SPROCKET TORQUE
8. SET MOTOR SELECTION CRITERIA
9. EVALUATE PUMP REQUIRED FOR EACH MOTOR
10. SELECT BEST PUMP/MOTOR COMBINATION

SEACO, INC.

TABLE 9

CALCULATION OF COUNTER ROTATION SPEED

FROM SHEET 3, MAXIMUM DIFFERENTIAL SPEED IS: 77.7 rpm

TRACK SPEED IS (77.7/2)*5.5 ft/rev = 213.7 fpm

VEHICLE ROTATION PERIOD IS: 2*PI*R/Vt

= 6.2832 * 4.4167 / 213.7

= 0.130 MINUTES

= 7.8 SECONDS

1/T = 7.7 REVOLUTIONS PER MINUTE

THE VEHICLE SLEW RATE IS MORE THAN 45 DEGREES PER SECOND

WHICH IS A HIGH SLEW RATE THAT IS PROBABLY ABOVE THE

CAPABILITY OF THE EXISTING VEHICLE/TRANSMISSION.

SEACO, INC.

TABLE 10 CANDIDATE VARIABLE DISPLACEMENT HYDRAULIC MOTORS FOR GSD-10

ITEM no.	MANUFACTURER	MODEL NUMBER	DISPL. in^3/rev	WEIGHT lbs.	SPEED rpm	PEAK TORQUE lb-ft
1	VON ROLL HYDRAULICS R.W. ATKINSON CO. 3713 SANTA FE AVE. LOS ANGELES, CA 90058 910-321-4001	MK-K-HS	11.36	255	2900 MAX	903 AT 6000 PSI
2	REXROTH CORPORATION P.O. BOX 2407 LEHIGH VALLEY, PA 18001 215 694-8300	A6V160	9.76	163	3500 at 7 deg 2650 at 25 deg	751 at 5800 psi
3	MARGE CARACIO	A6V225	13.73	227	3000 at 7 deg 2360 at 25 deg	1056 at 5800 psi
4	EATON CORPORATION	MV-76	7.60	228	2775	606
5	SPENCER DIVISION	MV-107	10.70	310	2480	848
6	32nd AVE. WEST SPENCER, IOWA 51301 712-264-3300	MV-149	14.90	420	2215	1187
7	COMMERCIAL SHEARING	BMV-140	8.51	147	3100	612
8	1775 LOGAN AVE.	BMV-186	11.36	165	2800	812
9	YOUNGSTOWN, OHIO 44501 216 098-2420	BMV-260	15.86	180	2500	1130
	SUNDSTRAND CORP.					
10	AMES, IOWA	MV-24	7.24	266	2700	480
11	50010	MV-25	10.12	370	2400	673
12	ABEX CORPORATION	MV-6	6	155	3000/3600	444
13	1220 DUBLIN ROAD	MV-7	7.25	155	3000/3600	537
14	COLUMBUS, OHIO 43216	MV-11	11	300	2400/2800	675
15	614 481-7300	MV-14	14	300	2400/2800	850

TABLE 11 - CANDIDATE FIXED-DISPLACEMENT HYDRAULIC MOTORS FOR GSD-10

ITEM no.	MANUFACTURER	MODEL NUMBER	DISPL. in^3/rev	WEIGHT lbs.	SPEED rpm	PEAK TORQUE lb-ft
16	VON ROLL HYDRAULICS	MF-K-HS-186	11.36	200	2900	903
17	R.W. ATKINSON CO.	MF-K-HS-92	5.51	110	3600	445
	3713 SANTA FE AVE.					
	LOS ANGELES, CA 90058					
	910-321-4001					
18	REXROTH CORPORATION	A2F-160	9.76	139	2650	751
	P.O. BOX 2407					at 5800 psi
	LEHIGH VALLEY, PA 18001					
	215 694-8300					
19	MARGE CARACIO	A2F-225	13.73	194	2360	1056
						at 5800 psi
20	EATON CORPORATION	MF-76	7.60	121	2275	606
21	SPENCER DIVISION	MF-107	10.70	160	2480	848
22	32nd AVE. WEST	MF-149	14.90	200	2215	1187
	SPENCER, IOWA 51301					
23	COMMERCIAL SHEARING	BMF-140	8.51	104	3100	612
24	1775 LOGAN AVE.	BMF-186	11.36	126	2800	812
	YOUNGSTOWN, OHIO 44501					
25	SUNDSTRAND CORP.	MF-36	6	95	4400	467
26	AMES, IOWA	MF-24	7.24	154	2700	480
27	50010	MF-27	10.12	175	2400	673
28	ABEX CORPORATION	MF-6	6	105	3000	444
29	1220 DUBLIN ROAD	MF-7	7.25	105	3000	537
30	COLUMBUS, OHIO 43216	MF-11	11	250	2400	675
31	614 481-7300	MF-14	14	250	2400	850
32	VOLVO HYDRAULICS	F11-110	6.72	102	3300	534
33	ROCKLEIGH IND. PARK	F11-150	9.15	155	3000	728
	ROCKLEIGH, NJ 07647					
	201 768-7300					

TABLE 12
COMPARISON OF VARIABLE DISPLACEMENT MOTORS

ITEM NO.	PEAK MOTOR SPEED rpm	REDUCTIC.. RATIO	PEAK MOTOR TORQUE lb-ft	PEAK SPROCKET TORQUE lb-ft	CORNER HORSE-POWER hp	MOTOR WEIGHT lbs	POWER TO WEIGHT RATIO hp/lb
1	2900	37.32303732	903	33702.70270	498.60048	255	1.955296
2	3500	45.04504504	751	33828.82882	500.4664	163	3.070346012
3	3000	38.61003861	1056	40772.20077	603.1872	227	2.657212334
4	2775	35.71428571	606	21642.85714	320.18616	228	1.404325263
5	2480	31.91763191	848	27066.15186	400.418816	310	1.2916736
6	2215	28.50707850	1187	33837.90218	500.600632	420	1.191906266
7	3100	39.89703989	612	24416.98841	361.22688	147	2.457325714
8	2800	36.03603603	812	29261.26126	432.89344	165	2.623596606
9	2500	32.17503217	1130	36357.78635	537.88	180	2.988222222
10	2700	34.74903474	480	16679.53668	246.7584	266	0.927663157
11	2400	30.88803088	673	20787.64478	307.53408	370	0.831173189
12	3600	46.33204633	444	20571.42857	304.33536	155	1.963453935
13	3600	46.33204633	537	24880.30888	368.08128	155	2.374717935
14	2800	36.03603603	675	24324.32432	359.856	300	1.19952
15	2800	36.03603603	850	30630.63063	453.152	300	1.510506666

TABLE 13
COMPARISON OF FIXED DISPLACEMENT MOTORS

ITEM NO.	PEAK MOTOR SPEED rpm	REDUCTION RATIO	PEAK MOTOR TORQUE lb-ft	PEAK SPROCKET TORQUE lb-ft	CORNER HORSE-POWER hp	MOTOR WEIGHT lbs	POWER TO WEIGHT RATIO hp/lb
16	2900	37.32303732	903	33702.70270	498.60048	200	2.4930024
17	3600	46.33204633	445	20617.76061	305.0208	110	2.772916363
18	2650	34.10553410	751	25613.25611	378.92456	139	2.726075971
19	2360	30.37323037	1056	32074.13127	474.507264	194	2.445913732
20	2775	35.71428571	606	21642.85714	320.18616	121	2.646166611
21	2480	31.91763191	848	27066.15186	400.418816	160	2.5026176
22	2215	28.50707850	1187	33837.90218	500.600632	200	2.50300316
23	3100	39.89703989	612	24416.98841	361.22688	104	3.473335384
24	2800	36.03603603	812	29261.26126	432.89344	126	3.435662222
25	4400	56.62805662	467	26445.30244	391.23392	95	4.118251789
26	2700	34.74903474	480	16679.53668	246.7584	154	1.602327272
27	2400	30.88803088	673	20787.64478	307.53408	175	1.7573376
28	3000	38.61003861	444	17142.85714	253.6128	105	2.41536
29	3000	38.61003861	537	20733.59073	306.7344	105	2.92128
30	2400	30.88803088	675	20849.42084	308.448	250	1.233792
31	2400	30.88803088	850	26254.82625	388.416	250	1.553664
32	3300	42.47104247	534	22679.53668	335.52288	102	3.28944
33	3000	38.61003861	728	28108.10810	415.8336	155	2.682797419

TABLE 14

GSD-10 - HYDRAULIC MOTOR SELECTION CRITERIA

GENERAL CRITERIA

1. LIGHT WEIGHT IS DESIRED - SET WEIGHT CUTOFF AT 175 lbs/motor.

2. CONTINUOUS OUTPUT POWER OF 150 hp OR GREATER

3. STEERING PERFORMANCE MUST BE BETTER THAN HS-400
   SET TE/Wt PER SPROCKET EQUAL TO .50 FOR STEER TORQUE.

   WEIGHT = 48,031.6 lbs.
   TE (PER SPROCKET) = 24,016 lbs.
   SPROCKET TORQUE = 21,014 ft-lbs.

THUS:

1. WEIGHT < 175 lbs.
2. HP > 150 (continuous)
3. SPROCKET TORQUE > 21,014 ft-lbs.

SEACO, INC.

TABLE 15
MOTORS MEETING THE THREE PERFORMANCE CRITERIA

ITEM NO.	PEAK MOTOR SPEED rpm	REDUCTION RATIO	PEAK MOTOR TORQUE lb-ft	PEAK SPROCKET TORQUE lb-ft	CORNER HORSE-POWER hp	MOTOR WEIGHT lbs	POWER TO WEIGHT RATIO hp/lb
2	3500	45.04504504	751	33828.82882	500.4664	163	3.070346012
7	3100	39.89703989	612	24416.98841	361.22688	147	2.457325714
8	2800	36.03603603	812	29261.26126	432.89344	165	2.623596606
13	3600	46.33204633	537	24880.30888	368.08128	155	2.374717935
18	2650	34.10553410	751	25613.25611	378.92456	139	2.726075971
21	2480	31.91763191	848	27066.15186	400.418816	160	2.5026176
23	3100	39.89703989	612	24416.98841	361.22688	104	3.473335384
24	2800	36.03603603	812	29261.26126	432.89344	126	3.435662222
25	4400	56.62805662	467	26445.30244	391.23392	95	4.118251789
32	3300	42.47104247	534	22679.53668	335.52288	102	3.28944
33	3000	38.61003861	728	28108.10810	415.8336	155	2.682797419

OF THE VARIABLE DISPLACEMENT UNITS, ITEM 2 WAS SELECTED
FOR FIXED DISPLACEMENT, NUMBER 32 WAS SELECTED
NOTE: NUMBER 32 HAS A HIGHER EFFICIENCY THAN NUMBER 25

TABLE 16
ANALYSIS OF PUMP REQUIREMENTS FOR EACH OF THE CANDIDATE MOTORS

MOTOR NO.	MAXIMUM MOTOR SPEED	MAX. MOTOR DISPLACEMENT cu-in/rev	MIN. MOTOR DISPLACEMENT cu-in/rev	OIL FLOW AT 20 mph SKIDOUT cu-in/min	DIFFERENTIAL TORQ CAPACITY AT SKIDOUT lb-ft	PUMP DISPL. FOR 1800 rpm ENGINE SPEED cu-in/rev
2	2900	9.76	2.81	8149	7556.3832	5.432666666
32	3300	6.72	6.72	22176	22678.98	14.784

THE VARIABLE DISPLACEMENT MOTOR HAS AN ASSOCIATED PUMP THAT IS SMALL BUT PRODUCES INSUFFICIENT TORQUE AT THE SKIDOUT CONDITION IF THE DISPLACEMENT IS ALLOWED TO GO TO THE MINIMUM (WHERE MAX. SPEED IS ALLOWED).

THE FIXED DISPLACEMENT MOTOR PROVIDES SUFFICIENT TORQUE, BUT REQUIRES A PUMP THAT IS LARGE.

THE USE OF A LARGE PUMP IS ACCEPTED SINCE IT MAY ALSO BE USED AS THE SOURCE OF POWER FOR THE WATERJET DRIVE THEREBY SIMPLIFYING THE 3-MODE CONTROL REQUIREMENTS

TABLE 17
SUMMARY OF WEIGHT AND CG RESULTS

OPTION 1

GSD-10 SYSTEM WEIGHT: 3611
VEHICLE WEIGHT: 49,090
GSD-10 SYSTEM CG: 105
VEHICLE CG: 194.3

OPTION 2

GSD-10 SYSTEM WEIGHT: 3600
VEHICLE WEIGHT: 48,870
GSD-10 SYSTEM CG: 108
VEHICLE CG: 195.1

OPTION 3

GSD-10 SYSTEM WEIGHT: 3424
VEHICLE WEIGHT: 48,903
GSD-10 SYSTEM CG: 149
VEHICLE CG: 199

OPTION 4

GSD-10 SYSTEM WEIGHT: 3311
VEHICLE WEIGHT: 48,730
GSD-10 SYSTEM CG: 113.3
VEHICLE CG: 196.75

TABLE 16

SUMMARY: STEERING COMPONENT SELECTION

STEER MOTOR:

VOLVO, FIXED DISPLACEMENT - 6.74 IN^3/REV

195 HP CAPABILITY

HYDRAULIC PUMP

ABEX, VARIABLE DISPLACEMENT FULL-OVER-CENTER

- 14 IN^3/REV
- 400 HP CAPABILITY

DISPLACEMENT CONTROL

ABEX SERVO VALVE WITH PUMP DISP. FEEDBACK

TABLE 19
POWER SINK SUMMARY - PEAK LOADS/MIN LOAD

COMPONENT	LAND MODE NO STEER	LAND MODE FULL STEER	WATER MODE	LAND/WATER MODE NO STEER	LAND/WATER MODE FULL STEER
ALTERNATOR a/	12/1	12/1	12/1	12/1	12/1
COOLING FAN b/	47	47	47	47	47
HYDRAULIC PUMP FOR AUXILIARIES	7/1	7/1	7/1	7/1	7/1
HYDRAULIC STEER/WJ DRIVE PUMP c/	---	150	---	---	150
AVAILABLE OUTPUT POWER					
TO TRANSMISSION AND TRACKS	334/351	184/201		75	75
TO WJ DRIVE			334/351	259/276	109/126

NOTES:

A) ALTERNATOR LOAD IS INTERMITTENT AS PER REQUIRED BY BATTTERIES AND VARIES WITH VEHICLE APPLICATION
B) POWER REQUIREMENT VARIES WITH ENGINE SPEED
C) THIS DENOTES POWER TO HYDRAULIC STEER/WJ DRIVE PUMP USED ONLY FOR AUXILIARY STEER MOTOR

TABLE 20

Effect of Speed and Transmission Gear Selection on Rolling Resistance Limitations

A. Proposed System

MSP-PEAK	825
N12	VARIES
N23	1.28
N34	3.05
NETA	0.70

← *Leyland Differential*

$N_{OVERALL} = 3.917$ in 5^{th}
$= 34.228$ in 1^{st}

GEAR	N12	SPROCKET TORQUE
1	7.973	18034.54
2	3.188	7211.102
3	3.021	6833.356
4	1.383	3128.279
5	1.000	2261.952

AERO DRAG

MPH	U FT/S	FD LB
30.000	44.000	238.902
35.000	51.333	325.173
40.000	58.667	424.715
45.000	66.000	537.530

Rolling resistance in lb/Ton GVW

GEAR	MAX SPEED	MAX RR
4	30	126.4698
4	35	123.1995
4	40	119.4261
4	45	115.1495
5	30	88.93804
5	35	85.66775
5	40	81.89433
5	45	77.61780

B. High-Torque / Low-Speed Option

MSP-PEAK	825
N12	VARIES
N23	5.43
N34	3.06
NETA	0.70

← *Dana Spicer; Note there is no trans-diff step-up*

- *Max torque increases*
- *Max speed decreases*

$N_{OVERALL} = 16.61$ in 5^{th}
$= 132.5$ in 1^{st}
!

GEAR	N12	SPROCKET TORQUE
1	7.973	76505.91
2	3.188	30590.85
3	3.021	28988.38
4	1.383	13270.74
5	1.000	9595.624

AERO DRAG

MPH	U FT/S	FD LB
30.000	44.000	238.902
35.000	51.333	325.173
40.000	58.667	424.715
45.000	66.000	537.530

GEAR	MAX SPEED	MAX RR
4	30	565.8706
4	35	562.6003
4	40	558.8269
4	45	554.5503
5	30	406.6537
5	35	403.3834
5	40	399.6100
5	45	395.3334

Vehicle weight: 26.38 tons

TABLE 21
GRADE CAPABILITY

```
      GRADE

     COEFFICIENT OF FRICTION    :      .7000
     ROLLING RESISTANCE, LB/LB :      .0400                 (0.06)

     MAXIMUM VEHICLE GRADE IS   :    33.1141 DEGREES         (32.2)
                                     65.2244 PERCENT GRADE   (62.9)
```

```
      ∇GRADE[⎕]∇
[0]   GRADE
[1]    ''
[2]    '      COEFFICIENT OF FRICTION    : ', 10 4 ⍕MU
[3]    '      ROLLING RESISTANCE, LB/LB : ', 10 4 ⍕F
[4]    ''
[5]    Z←NEWTON 0.5,1E¯4
[6]    G←3○Z
[7]    '      MAXIMUM VEHICLE GRADE IS   : ',(10 4 ⍕Z←Z×180÷○1),' DEGREES'
[8]    '                                   ',(10 4 ⍕G×100),' PERCENT GRADE'

      ∇ZERO[⎕]∇
[0]   Z←ZERO X
[1]    Z←((MU×2○X)-F)-1○X
```

TABLE 22

Summary of Key Parameters Used in TUTSIM Simulation of AAV7A1

Block	Variable	Value	Description
10	P_{ACC}	34,100 ft-lb/sec	Engine accessory power load (parasitic); degrades effective engine output by 62 Hp
12	T_{COMB}	VARIES	Engine torque due to combustion; varies as shown on Module P1
14	$\Delta\omega_{PM}$	26 rad/sec	Drop in engine speed associated with a gear shift
21	$\alpha_{PM, SET\ POINT}$	VARIES	Set point for angular acceleration of prime mover; simulates the time response of the prime mover to a given throttle setting.
22	$\omega_{PM}(0)$	167 rad/sec	Initial speed of prime mover
24	$\omega_{PM;MIN,MAX}$	150 rad/sec / 293 rad/sec	Prescribed limits to engine output speed
101	C_{SR}	[non-dim]	Ratio of the transmission turbine to prime mover speed.
102	C_{TR}	$f(C_{SR})$, [N-D]	Torque ratio of torque converter as a function of C_{SR}; look-up table; stall torque ratio of 3.03
103	$C_{TR;MIN,MAX}$	1.00-3.03	Min and max allowable torque ratios from torque converter.
105	T_{TURB}	—	Net torque of transmission turbine output, corrected for inertial loading of turbine during acceleration
106	ω_{TURB}	—	Angular velocity of turbine as given by integral of angular acceleration
122	η_{TRANS}	0.99	Mechanical efficiency of transmission unit
125	$J_{n_{TRANS}}$	65.9-1.3 lb-ft-s²	Inertia of transmission gear set as a function of engagement in gear 'n'; see module P2 layout for details
126	n	0 → 4	Index to denote gear engagement; 0 is 1st, 1 is 2nd, ...
127	ω_2	Constant / Variable	Angular velocity of transmission OUTPUT; is either constant as in AAV100, or variable as in AAV101.
150-4	N_{1-5}	7.973 - 1.00	Gear reduction ratio of gear 'n'
155-8	$\omega_{T_{1-4}}$	52.5 - 167.52	Transmission turbine angular velocity shift thresholds
203	η_{SD}	0.99	Mechanical efficiency of steer drive in rectilinear motion
204	J_{23}	49.0 lb-ft-s²	Effective moment of inertia for Leyland differential unit
205	N_{23}	1.28	Gear reduction ratio for Leyland differential
303	η_{FD}	0.99	Mechanical efficiency of final drive units
304	J_{34}	72.1 lb-ft-s²	Effective moment of inertia for final drive unit
305	N_{34}	3.06	Gear reduction ratio for final drive
405	R	0.875	Radius of drive sprocket
403	W/g	1639 $\frac{lb_f \cdot s^2}{ft}$	Mass of vehicle
408	J_{SP}	100	Effective inertia of sprocket, including tracks and road wheels

TABLE 23
APPROXIMATION FOR INERTIA FOR DANA-SPICER DIFFE[illegible]TIAL

INERTIA

GEAR TRAIN EQUIVALENT INERTIA COMPUTATION ROUTINE

(C)REATE OR (E)XECUTE DATA MATRIX ?
E

*** PROCESS DATA ***

ASSEMBLY NUMBER 1 HAS 3 COMPONENTS

COMP NUMB	TYPE	RADIUS IN	LENGTH IN	MATERIAL LB/IN3	N[I,J]	INTERFACE ASSY	INTERFACE COMP	INERTIA LB-F-S2	I/NIJ*2 LB-F-S2
1	G	1.50	1.50	.2836	1.000	0	0	2.82E-001	
2	S	.88	6.00	.2836	1.000	0	0	1.31E-001	
3	G	1.75	1.75	.2836	.350	2	1	6.09E-001	
								1.02E0000	8.34E0000

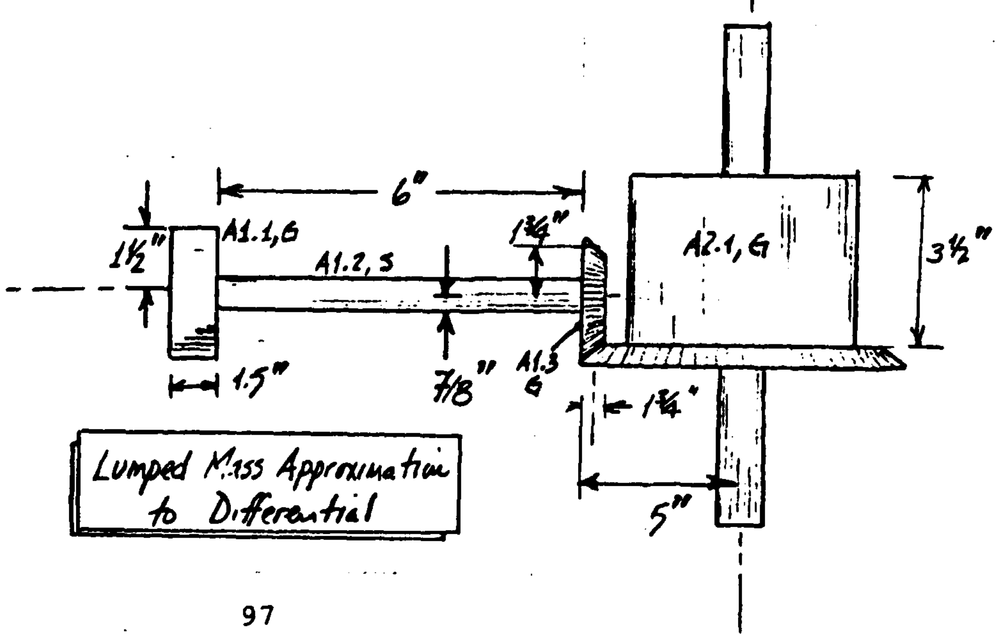
ASSEMBLY NUMBER 2 HAS 1 COMPONENTS

COMP NUMB	TYPE	RADIUS IN	LENGTH IN	MATERIAL LB/IN3	N[I,J]	INTERFACE ASSY	INTERFACE COMP	INERTIA LB-F-S2	I/NIJ*2 LB-F-S2
1	G	5.00	3.50	.2836	2.857	1	3	8.12E0001	
								8.12E0001	9.95E0000

Approximation to Dana-Spicer IC-44

*** END OF JOB ***

$I_{Eff} = 1.02 + 9.95 = 10.97$ lb-ft-s^2



Lumped Mass Approximation to Differential

TABLE 24
APPROXIMATION FOR INERTIA FOR LEYLAND DIFFERENTIAL

LEYLAND∆B

0	0	0	0	0	0	0	0	0	0
1	1	2	1.5	1.5	1	0	0	0	0
1	2	1	0.875	6	1	0	0	0	0
1	3	2	3.9	1.25	1	1	2	1	0
2	1	2	5	1.25	1	1	1	3	0
2	2	1	4	3.5	1	0	0	0	0

INERTIA

GEAR TRAIN EQUIVALENT INERTIA COMPUTATION ROUTINE

(C)REATE OR (E)XECUTE DATA MATRIX ?
E

*** PROCESS DATA ***

ASSEMBLY NUMBER 1 HAS 3 COMPONENTS

COMP NUMB	TYPE	RADIUS IN	LENGTH IN	MATERIAL LB/IN3	N[I,J]	INTERFACE ASSY	COMP	INERTIA LB-F-S2	I/NIJ*2 LB-F-S2
1	G	1.50	1.50	.2836	1.000	0	0	2.82E¯001	
2	S	.88	6.00	.2836	1.000	0	0	1.31E¯001	
3	G	3.90	1.25	.2836	.780	2	1	1.07E0001	

								1.11E0001	1.83E0001

ASSEMBLY NUMBER 2 HAS 2 COMPONENTS

COMP NUMB	TYPE	RADIUS IN	LENGTH IN	MATERIAL LB/IN3	N[I,J]	INTERFACE ASSY	COMP	INERTIA LB-F-S2	I/NIJ*2 LB-F-S2
1	G	5.00	1.25	.2836	1.282	1	3	2.90E0001	
2	S	4.00	3.50	.2836	1.000	0	0	3.33E0001	

								6.23E0001	3.79E0001

*** END OF JOB ***

Approximation to Leyland-B differential with 1.28 reduction ratio

$I_{EFF} = 11.1 + 37.9 = 49 \ lb\text{-}ft\text{-}s^2$

TABLE 25
APPROXIMATION FOR INERTIA FOR FINAL DRIVE UNIT

```
     FINAL△DRIVE
0    0    0    0     0     0    0    0    0    0
1    1    2    3.5   1.25  1    1    2    1    0
2    1    2    10.7  1.25  1    1    1    1    0

     INERTIA

    GEAR TRAIN EQUIVALENT INERTIA COMPUTATION ROUTINE

    (C)REATE OR (E)XECUTE DATA MATRIX ?
                  E

         *** PROCESS DATA ***
```

ASSEMBLY NUMBER 1 HAS 1 COMPONENT

COMP NUMB	TYPE	RADIUS IN	LENGTH IN	MATERIAL LB/IN3	N[I,J]	INTERFACE ASSY	COMP	INERTIA LB-F-S2	I/NIJ*2 LB-F-S2
1	G	3.50	1.25	.2836	.327	2	1	6.96E0000	

								6.96E0000	6.51E0001

ASSEMBLY NUMBER 2 HAS 1 COMPONENT

COMP NUMB	TYPE	RADIUS IN	LENGTH IN	MATERIAL LB/IN3	N[I,J]	INTERFACE ASSY	COMP	INERTIA LB-F-S2	I/NIJ*2 LB-F-S2
1	G	10.70	1.25	.2836	3.057	1	1	6.08E0002	

								6.08E0002	6.51E0001

*** END OF JOB ***

Approximation to final drive inertia

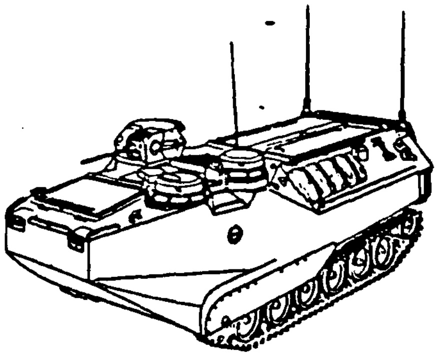
$$I_{EFF} = 6.96 + 65.1 = 72.1 \ \text{lb}\cdot\text{ft}\cdot\text{s}^2$$

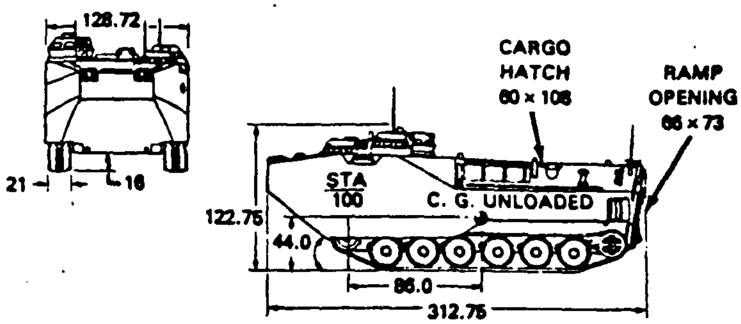
APPENDIX A - LVTP7A1 CHARACTERISTICS

APPENDIX A-1. TECHNICAL DATA SHEET FOR LVTP7A1

TECHNICAL DATA SHEET

VEHICLE: LVTP7A1
TYPE: LANDING VEHICLE, TRACKED, PERSONNEL, MODEL 7A1 (Assault Amphibian Personnel and Cargo Carrier)





1. **GENERAL**
   Weight (Cargo Loaded): 52,770 Pounds
   Crew: 3
   Weight Unloaded (Less Crew and Fuel): 40,932 Pounds
   Maximum Load Capacity: 25 Combat Equipped Troops or 10,000 Pounds of Cargo
   Center of Gravity:
   Unloaded: 44.0 Inches Above Ground, 86.0 Inches from Station 100.0
   Loaded: 42.6 Inches Above Ground, 98.6 Inches from Station 100.0
   Maximum Sea Water Draft (Cargo Loaded): 68.7 Inches
   Freeboard to Driver's Hatch Coaming: 33.6 Inches
   Freeboard to Inlet of Air Aspirator Valve: 35.8 Inches
   Unit Ground Pressure (Cargo Loaded, Zero Penetration): 8.0 PSI
   Fuel Capacity: 171 Gallons

2. **PERFORMANCE**
   Gross Horsepower to Weight Ratio: 15.2 HP/Ton
   Net Horsepower to Weight Ratio: 10.1 HP/Ton
   Drawbar Pull (Maximum at Stall Tractive Effort): 42,216 Pounds On Level Firm Terrain
   Cruising Range:
   Land at 25 MPH: 300 Miles
   Water at 2,600 RPM: 7 Hours
   Cruising Speed:
   Land: 20 to 30 MPH
   Water: 8 MPH
   Maximum Speed Forward:
   Land: 45 MPH
   Water: 8.4 MPH
   Maximum Speed Reverse:
   Land: 12 MPH
   Water: 4.5 MPH
   Obstacle Ability: 8-Foot Trench Span, 3-Foot Vertical Wall
   Maximum Forward Grade (Cargo Loaded): 60%
   Maximum Side Slope (Cargo Loaded): 60%
   Ground Clearance (Cargo Loaded): 16 Inches
   Minimum Turning Radius:
   Land: Pivot
   Water: Pivot
   Surf Ability: 10-Foot Plunging Surf Going Both Seaward and to Shore

3. **ENGINE**
   Make: Cummins
   Model: VT400
   Type: 4 Cycle 8 Cylinder, 90° Vee, Water Cooled, Turbocharged
   Bore: 5.5 Inches
   Stroke: 4.75 Inches
   Displacement: 903 Cubic Inches
   Compression Ratio: 15.5:1
   Fuel: Multifuel
   Rated Horsepower: 400 ± 5% at 2800 RPM with DF-2
   Rated Torque: 825 ft-lbs ± 5% at 2050 RPM with DF-2
   Oil Capacity: 9.4 Gallons
   Coolant System Capacity: 30 Gallons

4. **POWER TRAIN**
   Transmission: NavSea HS-400-3A1
   Type: Hydraulic Torque Converter, Parallel Shaft Gear Arrangement
   Maximum Converter Torque Multiplication: 2.83:1
   Gear Ratios, Forward:
   First Speed: 8.27:1
   Second Speed: 4.63:1
   Third Speed: 2.25:1
   Fourth Speed: 1.27:1
   Reverse Uses First and Second Speed Ratios
   Final Drive Ratio: 3.06:1
   Overall Maximum Torque Ratio (Engine to Sprocket): 70.8:1
   Transmission Oil Capacity: 23 Gallons (with Oil Coolers, Filters, Lines)

5. **RUNNING GEAR**
   Type: Torsion Bar and Tube Suspension, Front Sprocket, Raised Rear Idler
   Number of Wheels: 6 Rubber Tired Dual per Side, 26 Inch Diameter
   Number of Return Idlers: 1 Per Side, 20 Inch Diameter Wheels
   Sprocket:
   Number of Teeth: 11
   Feet Per Revolution: 5.5
   Track:
   Type: Steel, Single Pin, Rubber Bushed, With Replaceable Pads
   Number of Shock Absorbers: 3 Per Side

Number of Blocks: 84 Per Side
Pitch: 6 Inches
Weight Per Block: 31.4 Pounds
Weight Per Side: 2638 Pounds

6. WATER PROPULSION
Water Jet Pumps
Capacity: 14,000 GPM
Thrust: 3,025 Pounds Static
Quantity: 2
Location: Port and Starboard, aft
Steering and Reverse by Jet Deflectors

7. ELECTRICAL
Nominal Voltage: 24
Generator: 300 AMP
Battery:
Volts: 12
Type: 6TN
Quantity: 4

8. COMMUNICATIONS
Radio:

RT-246/VRC	Receiver-Transmitter:	1
R-442/VRC	Receiver:	2
RT-524/VRC	Receiver-Transmitter:	1

Crypto Equipment:

TSEC/KY-57	Voice Security Set:	4

Location: Personnel Compartment, Starboard Side
5-Station Intercom System

9. ARMOR
Permanent Hull: Aluminum Armor Plate
Ramp Outer: 1.000 Inch
Ramp Inner: .500 Inch
Sides: 1.750, 1.395 and 1.222 Inches
Top: 1.185 Inches
Bottom: 1.185 Inches
Stern: 1.395 Inches

10. FIRE EXTINGUISHERS
Fixed:
Number of Cylinders: 1
Location: Port Sponson
Capacity: 17 Pounds Halon 1301
Portable:
Number of Cylinders 1
Location: Port Stanchion
Capacity: 2-3/4 Pounds Halon 1301
Automatic Fire Detection and Suppression System:
Number of Cylinders: 3 in Personnel Compartment
1 in Engine Compartment
Location: Aft Engine Compartment Bulkhead and Starboard Stanchion
Capacity: 7 Pounds Each Halon 1301

11. VISION AND SIGHTING EQUIPMENT
Driver's Station:
Direct Vision Blocks: 7
Driver's Night Vision Device, AN/VVS-2 1
Commander's Station:
Direct Vision Blocks: 7
Periscope, M27: 1
Armament Station:
Direct Vision Blocks: 9
Direct Ring Sight: 1
Indirect Optional Sight:
8 × Power NavSea P/N 2588628
1 × Power with Projected Reticle
Ramp:
Direct Vision Block: 1

12. CARGO COMPARTMENT
Length: 13.5 Feet
Width: 6.0 Feet
Height: 5.5 Feet
Volume: 445.5 Cubic Feet
Capacity: 24 Combat Equipped Troops

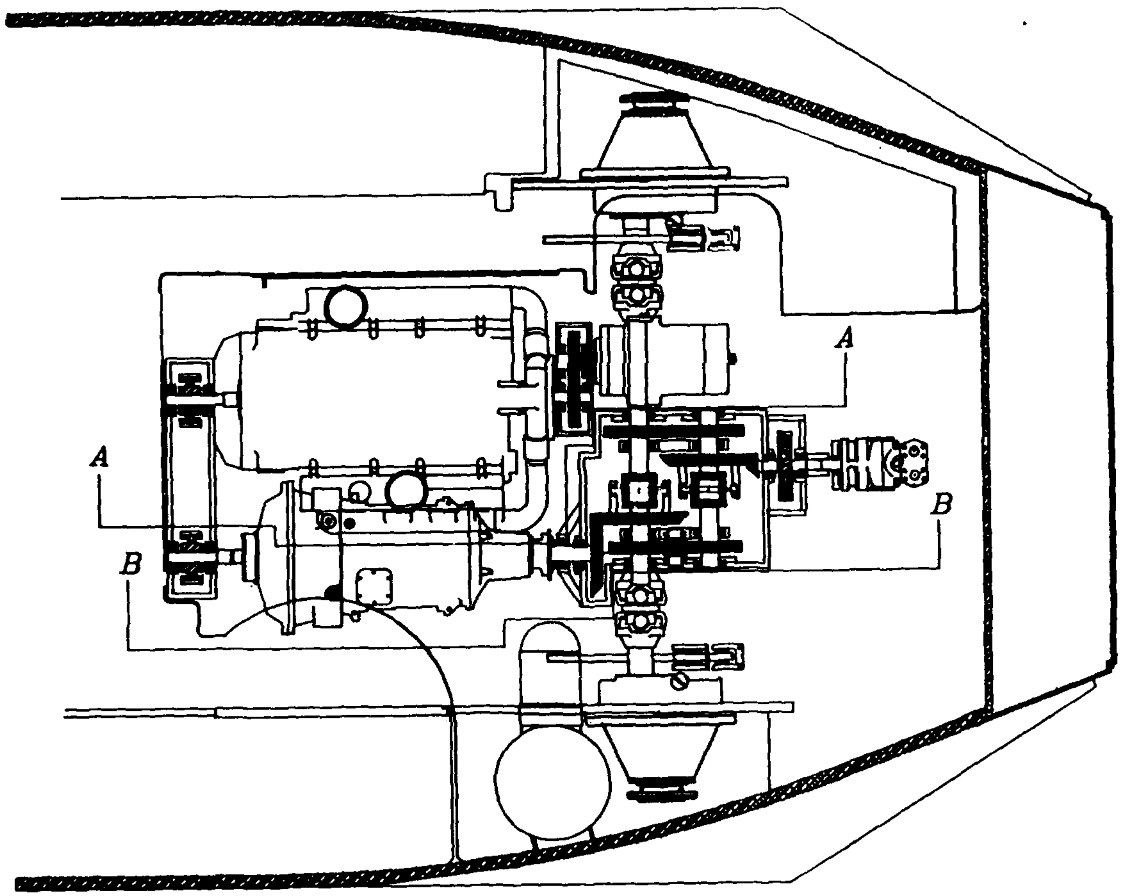
13. ARMAMENT AND AMMUNITION
Caliber .50 Machine Gun, M85
Traverse: 360 Degrees
Elevation: 60 Degrees
Depression: 15 Degrees
Power Control System: Electric-Manual
Ammunition: Caliber .50, 400 Ready Rounds
Rate of Fire (Cyclic):
High 1050 Rds/Min
Low 450 Rds/Min
Muzzle Velocity 2840 FPS
Range (Maximum) 7275 YDS

14. NAVIGATION EQUIPMENT
Magnetic Navigation Set

15. OTHER
Accessory Equipment:
Visor Kit, NavSea P/N 2587015
Litter Kit, NavSea P/N 2600077
Winterization Kit, NavSea P/N 2600063
Sight Kit Installation, Night Vision, NavSea P/N 2588618
Reference Data and Literature:
Vehicle Operation TM-07007A-10A and Supplement for LVTP7A1
Vehicle Maintenance TM-07007A-25A and Supplement for LVTP7A1

Contractor:
FMC Corporation
Date First Prototype: 1979

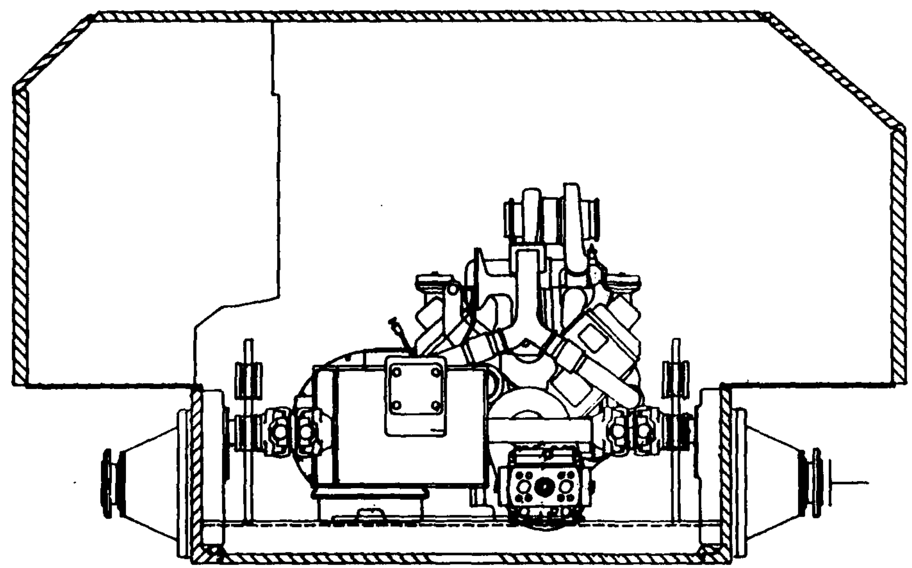
Appendix D



REVISIONS

SEACO, INCORPORATED
GSD-10, PLAN VIEW IN AAV-7A1

SCALE: 1"= 8"	REVISION: 26 FEB 87 BY: P.BODE & R.SCOTT
DATE: 11/30/86	DRAWN BY: MSR



REVISIONS

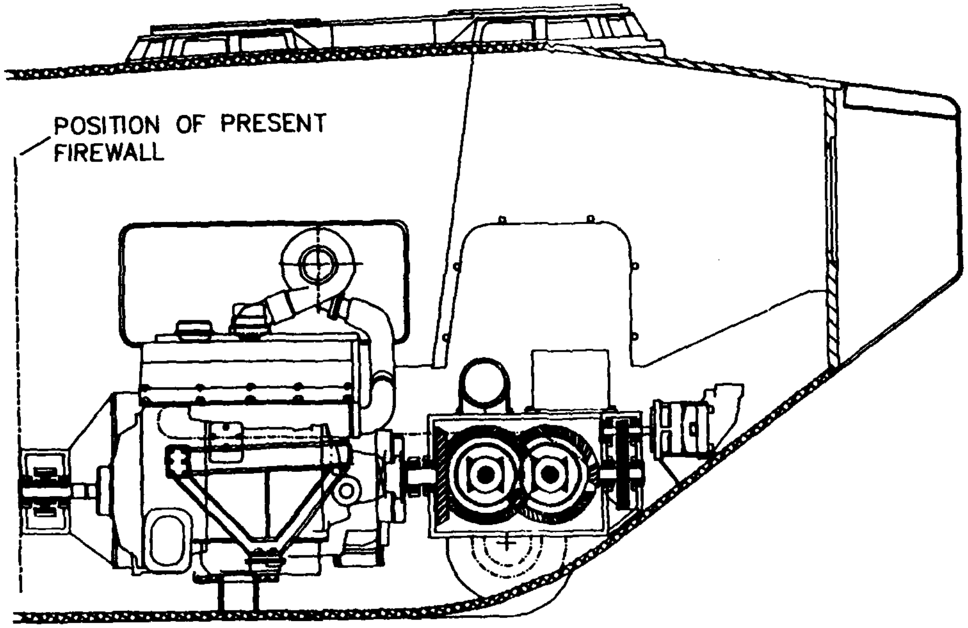
SEACO, INCORPORATED

GSD-10, END VIEW IN AAV-7A1

SCALE: 1" = 8"	REVISION: BY: P.BODE & R.SCOTT
DRAWN: 28 FEB 87	DRAWN BY : P.BODE & R.SCOTT

NOTES:

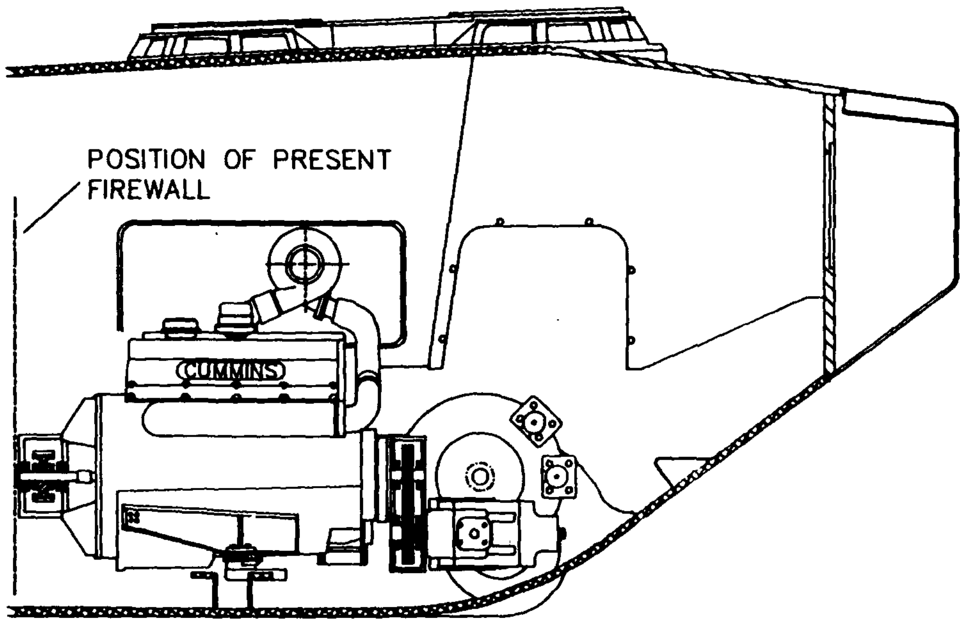
HYDRAULIC PUMP, HYDRAULIC PUMP TRANSFER CASE, AND BRAKES OMITTED FOR CLARITY



SECTION B–B
ROTATED 90 deg CCW

REVISIONS	

SEACO, INCORPORATED GSD-10, SIDE VIEW IN AAV-7A1	
SCALE: 1" = 8"	REVISION: 28 FEB 87 BY: P.BODE & R.SCOTT
DATE: 12/30/88	DRAWN BY : P.BODE & R.SCOTT



SECTION A-A
ROTATED 90 deg CCW

REVISIONS	

SEACO, INCORPORATED GSD-10, SIDE A VIEW IN AAV-7A1	
SCALE: 1" = 8'	REVISION: 28 FEB 87 BY: P.BODE & R.SCOTT
DRAWN 5 JAN 87	DRAWN BY : P.BODE & R.SCOTT